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# SOLAR ENERGY SYSTEM PERFORMANCE EVALUATION - FINAL REPORT FOR HONEYWELL OTS 41 SHENANDOAH (NEWNAN), GEORGIA


Prepared by

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Technology Strategy Center
1700 W. Highway 36
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Under Contract NAS8-32093/DE-AC03-81CS 30574

National Aeronautics and Space Administration
George C. Marshall Space Flight Center, Alabama 35812

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For the U.S. Department of Energy


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16. ABSTRACT

This report has been developed for the Department of Energy as part of the Solar Heating and Cooling Development Program. It is one of a series of reports describing the operational and thermal performance of a variety of solar systems installed in Operational Test Sites under this program. It describes the operation and technical performance of the Solar Operational Test Site (OTS 41) located at Shenandoah, Georgia, based on the analysis of data collected between January and August 1981. The following topics are discussed: system description, performance assessment, operating energy, energy savings, system maintenance, and conclusions. The solar energy system at OTS 41 is a hydronic heating and cooling system consisting of 702 square feet of liquid-cooled flat-plate collectors; a 1000-gallon thermal storage tank; a 3-ton capacity organic Rankine-cycle-engine-assisted air conditioner; a water-to-air heat exchanger for solar space heating; a finned-tube coil immersed in the storage tank to preheat water for a gas-fired hot water heater; and associated piping, pumps, valves, and controls. The solar system has six basic modes of operation and several combination modes. The system operation is controlled automatically by a Honeywell-designed microprocessor-based control system, which also provides diagnostics. Based on the instrumented test data monitored and collected during the 7 months of the Operational Test Period, the solar system collected 53 MMBtu of thermal energy of the total incident solar energy of 219 MMBtu and provided 11.4 MMBtu for cooling, 8.6 MMBtu for heating, and 8.1 MMBtu for domestic hot water. The projected net annual energy savings due to the solar system were approximately 50 MMBtu of fossil energy (49,300 cubic feet of natural gas) and a loss of 280 kWh(e) of electrical energy.

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# FOREWORD

This report has been developed for the Department of Energy as a part of the Solar Heating and Cooling Development Program. It describes the operation and technical performance of the Solar Operational Test Site (OTS) at Shenandoah (Newnan), Georgia and evaluates how the OTS has been functioning throughout a specified period of time, based on the analysis of data collected by the Site Data Acquisition System (SDAS). The objectives of the analysis are to report the long-term performance of the installed system and to make technical contributions to the definition of techniques and requirements for solar energy system design.

The contents of this document have been divided into the following topics of discussion:

- System Description.
- Performance Assessment.
- Operating Energy.
- Energy Savings.
- Maintenance.
- Summary and Conclusions.

Data used for the analysis have been collected, processed and maintained under the National Solar Data Network program, and were the major inputs used to perform the long-term technical assessment. The data have been archived by Automation Industries Inc., Vitro Laboratories Division, Silver Springs, Maryland.

## SYSTEM HISTORY

In July 1976, Honeywell Energy Resources Center (Minneapolis, Minnesota) entered into a contractual agreement with NASA's Marshall Space Flight Center to design and develop solar-powered building space heating and cooling systems. This on-going engineering field test effort is known as the "404" program. The objective of the program was to develop, install and test solar heating and cooling systems that (1) exhibit efficient performance capabilities, (2) were low in cost, and (3) were modular in composition to enhance application.

Honeywell was the prime contractor for the program team. Barber-Nichols Engineering of Denver, Colorado, and Lennox Industries of Marshalltown, Iowa, were subcontractors. Honeywell was responsible for the solar system design, overall program management and subcontractor coordination. Barber-Nichols and Lennox worked as a team to develop solar-powered Rankine engine/air conditioner subsystems. Lennox Industries supplied HVAC products suitable for application in the system, including the production flat-plate solar collector. Data acquisition and reduction was done by IBM, Inc., and Vitro Laboratories.

A single family residence, located in Shenandoah near Newman, Georgia, was selected as Operational Test Site 41 for this program. Installation of the system was begun in mid-July 1980. Debugging of the system was completed in mid-November. The Installation Acceptance Review (IAR) was held on-site on December 4, 1980, upon which commenced the Operational Test Period (OTP). The construction of the house was completed in early January 1981. The house has been occupied by a family of 4 since January 14, 1981.

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# SYSTEM DESCRIPTION

The Honeywell Shenandoah solar heating and cooling system is located in a single-family residence near Newnan, Georgia. The system is a double-loop, solar-assisted, hydronic heating and cooling system with a separate water storage and domestic water heating loop. Central heating is provided by stored thermal energy released through a water heating coil installed in the return air duct of a conventional gas-fired furnace. Central cooling is provided by a solar-powered Rankine-engine/auxiliary-electric-motor-driven air conditioner system (RC/AC).

The system (see Figures 1-1 through 1-3) consists of a solar collector primary loop that interfaces with a secondary water storage loop through a tube-and-shell heat exchanger. A solution of 40 percent ethylene glycol (antifreeze) in water is circulated through the primary loop, and plain water with a corrosion inhibitor is circulated through the secondary loop. The glycol/water collector loop consists of solar collectors installed on the roof of the residence, circulating pumps and hydronic specialties, which are contained within the energy transport module (ETM), the Rankine-cycle engine and control valves. The collector array consists of 702 square feet (gross area) of roof-mounted flat plate collectors facing 10 degrees west of due south at a tilt of 30 degrees from horizontal. The secondary loop consists of a 1000-gallon storage tank, circulating pumps and the tube side of the heat exchanger (contained within the ETM), and the solar heating coil. For reference purposes, a simplified wiring layout has been included (Figure 1-3). Domestic hot water (DHW) preheating is accomplished by a preheat coil immersed inside the storage tank.

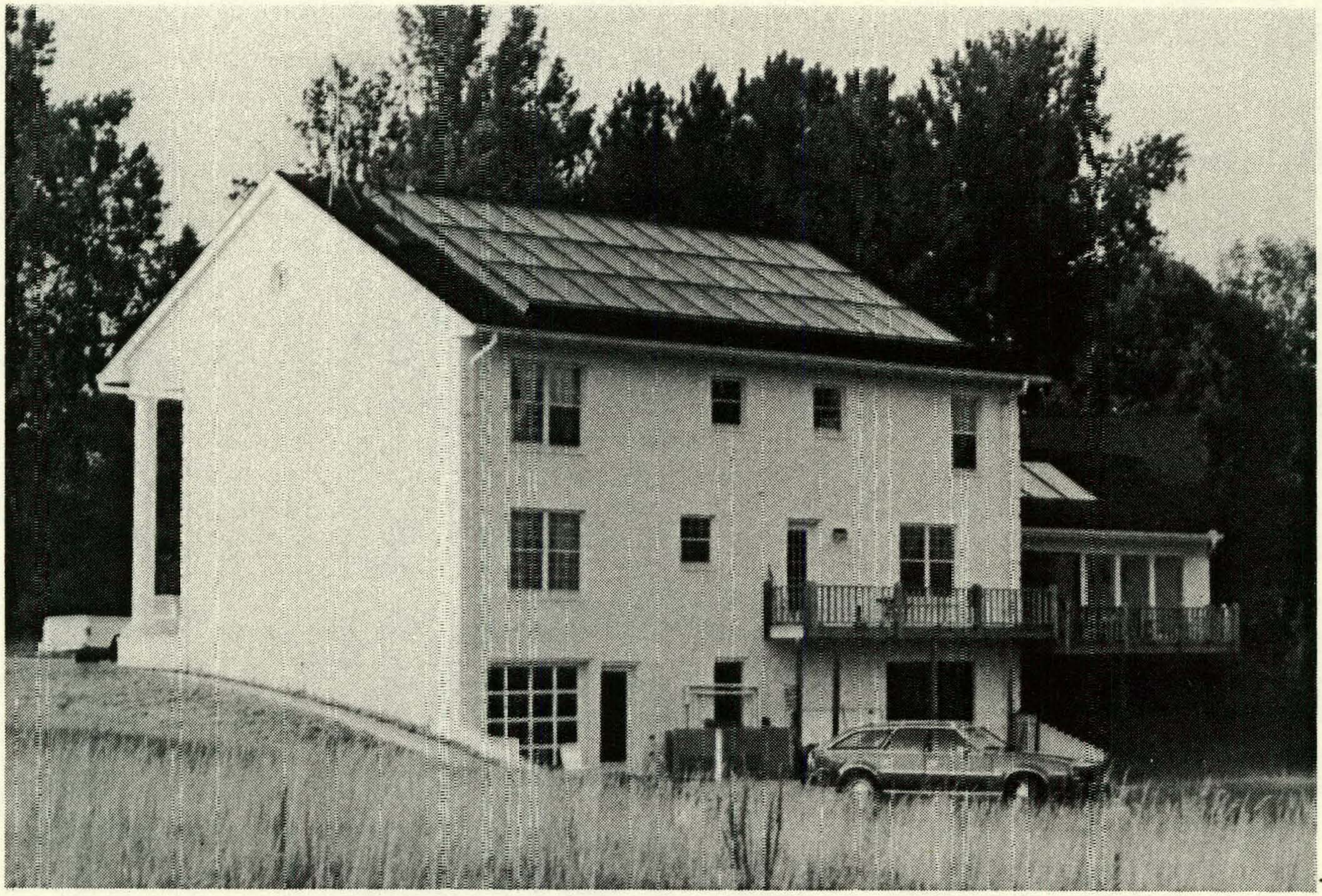


FIGURE 1-1. SHENANDOAH ROOF-MOUNTED COLLECTOR ARRAY

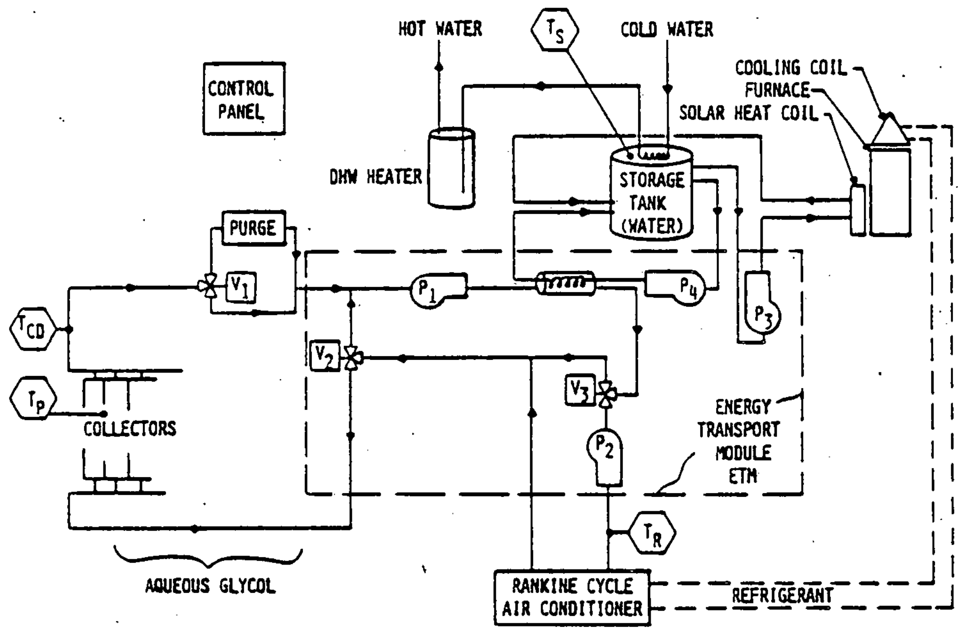



FIGURE 1-2. SHENANDOAH SINGLE-FAMILY RESIDENCE (SFRH/C) SYSTEM SCHEMATIC

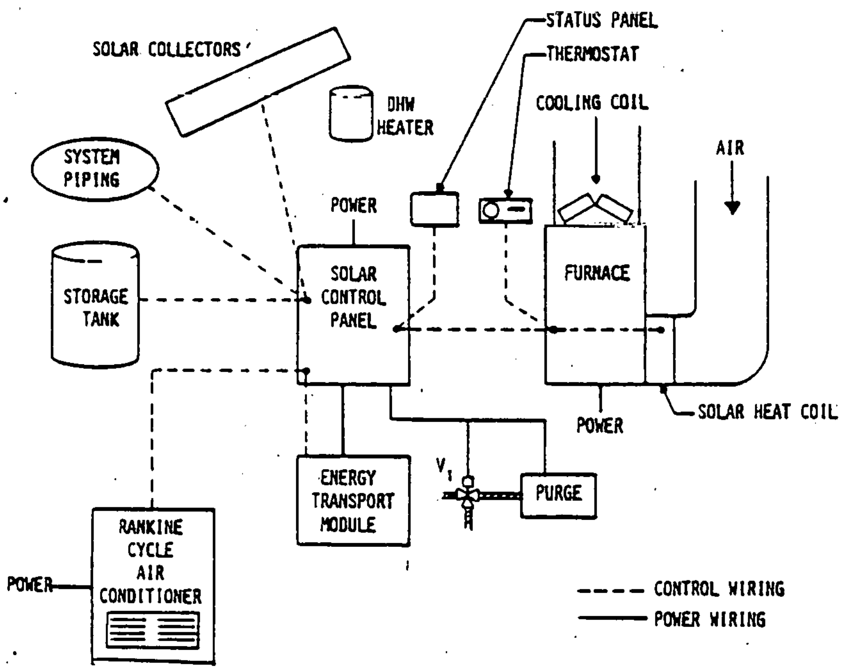



FIGURE 1-3. MODEL 404 SFRH/C CONTROLS/WIRING LAYOUT

## 1.1 SYSTEM OPERATING MODES

The system provides total space conditioning for a single-family residence, controlled through the space thermostat, using available solar energy in conjunction with conventional heating, ventilating, and air conditioning (HVAC) systems. First-stage heating uses solar energy from storage and first-stage cooling uses solar energy directly from the collectors or from solar storage. Second-stage conditioning is provided by the furnace (or electrically driven RC/AC). Solar energy not required for space conditioning is used by the Rankine engine to generate electricity or is rejected through a purge coil. Sizing and control of the conventional (sometimes referred to as "auxiliary") heating and cooling equipment is such that space conditioning is virtually uninterrupted during solar system servicing. The system has six operating modes.

### 1.1.1 Heating Mode

The space heating subsystem is activated when the space thermostat calls for heat. The solar system is configured such that energy is transferred from storage to the conditioned space, independent of the collector loop (Figure 1-4). When first-stage (solar) demand exists and energy is available in storage, pump $P_3$ is activated to provide hot storage water to the solar heat coil in the return air plenum. This occurs as long as $T_S$ (storage tank temperature) is greater than $110^{o}F$. (NOTE: All listed temperatures are adjustable.)

When solar energy is available and storage needs recharging, energy is transferred from the collectors to storage (see Subsection 1.1.2, Storage Charging Mode). When storage is depleted or when the heat coil cannot supply the full load, second-stage (auxiliary) heating is supplied by the furnace through electrical resistance heating elements.

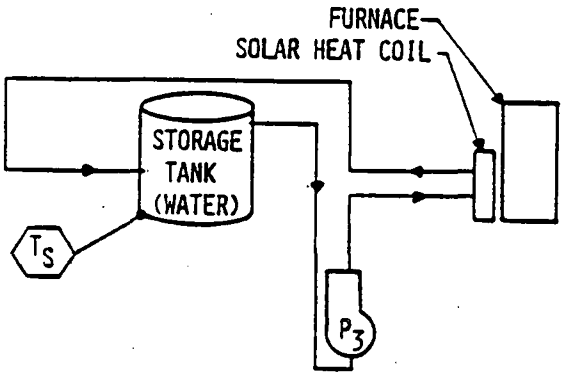



Figure 1-4. Heating Mode Schematic

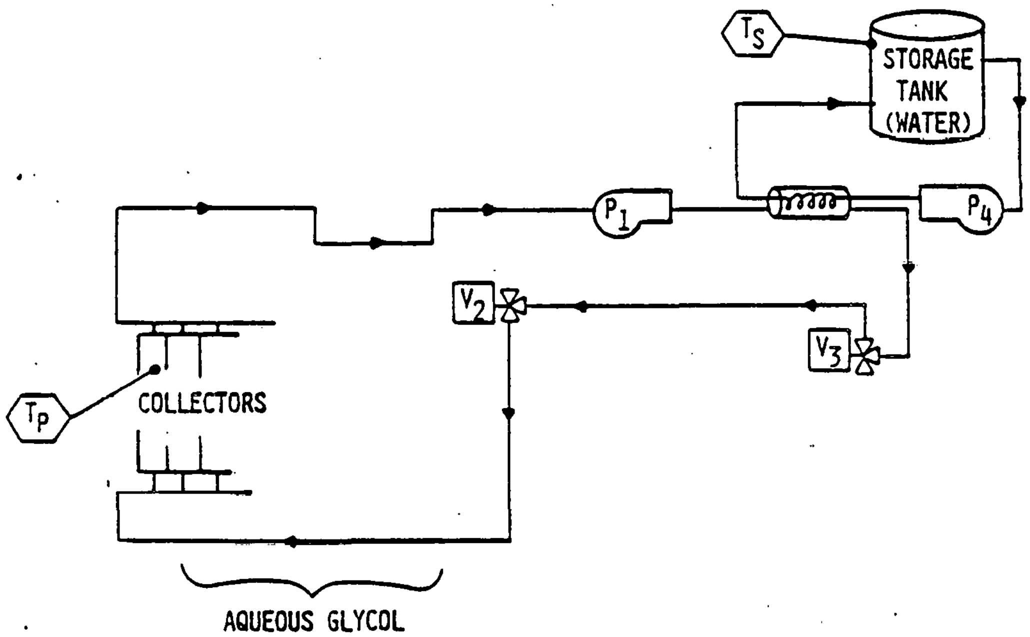



Figure 1-5. Storage Charging Mode Schematic

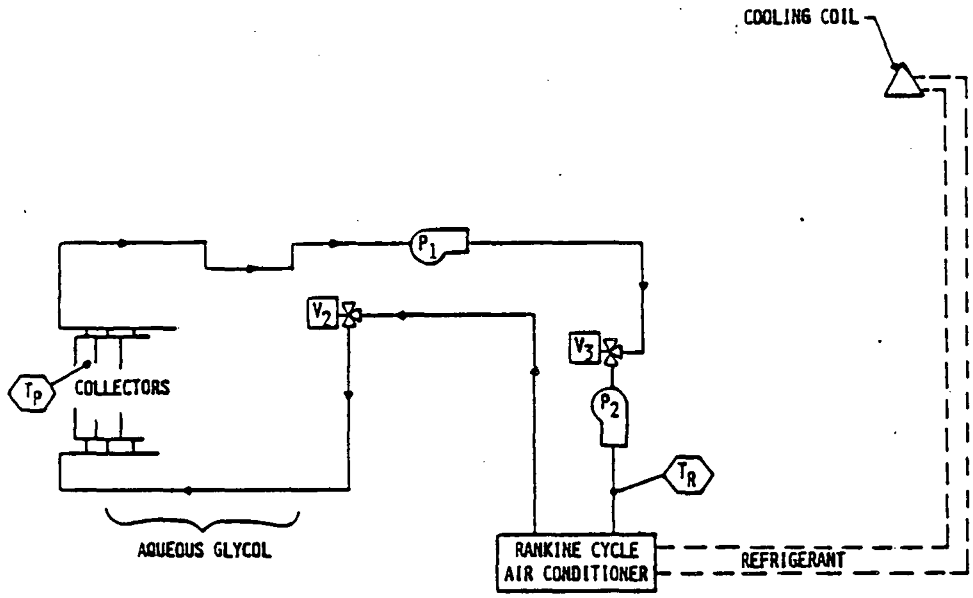
### 1.1.2 Storage Charging Mode

In any thermostat mode, when there is no demand for air conditioning, available solar energy will be transferred from the collectors to storage if the temperature difference between the collector plate, $T_p$, and the storage tank, $T_S$, is greater than 18F$^o$ (Figure 1-5). Pumps $P_1$ and $P_4$ are activated. Valve $V_3$ diverts flow around the RC/AC and Valve $V_2$ directs flow through the collectors. Charging continues until the storage tank temperature reaches the generation set point (160$^o$F with the thermostat in "OFF" or "HEAT" mode; 200$^o$F in "COOL" mode) or until the temperature difference, $T_p - T_S$, is less than 3F$^o$.

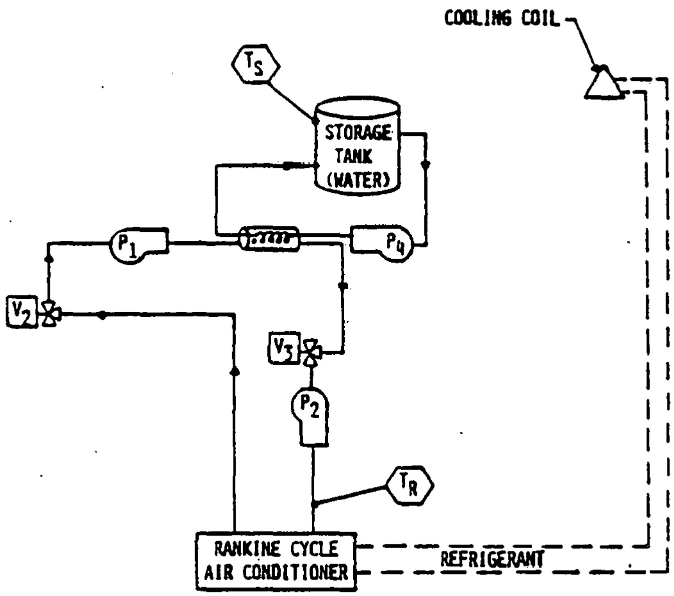
### 1.1.3 Cooling Mode

When the space thermostat calls for cooling, the RC/AC electric motor is activated to drive the compressor, which then supplies refrigerant to the direct expansion evaporator coil. When the compressor is running and solar energy is available, either directly from the collectors or indirectly from storage, the Rankine-cycle turbine (on a common shaft with the motor) is activated and unloads the motor. Figure 1-6 is a schematic of the cooling mode, both direct cooling and cooling from storage.

1.1.3.1 Cooling, Direct and from Storage--A call for cooling activates the furnace fan and the electrically driven air conditioning compressor. If $T_p$ (collector plate temperature) is greater than 165$^o$F, pump $P_1$ is activated and valves $V_3$ and $V_2$ are positioned to direct flow through the Rankine-cycle engine and through the collectors, respectively. When $T_p$ is less than 165$^o$F and $T_S$ (storage tank temperature) is greater than 170$^o$F, pump $P_1$ is activated and valves are positioned to direct flow through the Rankine cycle and to divert flow around the collectors. In addition, pump $P_4$ is activated to discharge storage energy into the RC/AC loop. When $T_R$ (RC supply temperature) reaches 160$^o$F, the Rankine cycle is activated to unload the motor. Pump $P_2$ also is activated to provide adequate glycol flow and heat transfer.




a. Direct Cooling




b. Cooling from Storage

Figure 1-6. Cooling Mode Schematic

1.1.3.2 Auxiliary Cooling--If adequate energy is not available from the collectors or from storage, the electric motor continues to drive the compressor to provide cooling.

### 1.1.4 Generation Mode

When there is no instantaneous demand for heating or cooling and the storage tank is fully charged (greater than 160°F in the "HEAT" or "OFF" mode; greater than 200°F in the "COOL" mode), the RC/AC will use excess energy to generate electricity (Figure 1-7). When the above conditions are satisfied and $T_p$ is greater than 195°F, pumps $P_1$ and $P_2$ are activated and valves $V_3$ and $V_2$ divert flow through the RC/AC and the collectors. Then, when $T_R$ reaches 190°F, the generation mode of the RC/AC is activated. The compressor clutch is disengaged, and the Rankine-cycle turbine is activated to drive the motor/generator. Note that the generation mode will not occur when the RC is disabled for repair or during winter shutdown.

### 1.1.5 Purge Mode

For overtemperature protection, valve $V_1$ diverts flow through the purge unit, and the purge fan is energized if the collector loop temperature, as sensed by $T_{CD}$, is greater than 220°F (Figure 1-8). This mode is available simultaneously with any other mode of operation.

### 1.1.6 Domestic Hot Water Preheating

The cold water supply to the domestic water heater is preheated as it passes through finned copper tubing submerged in the storage tank (Figure 1-9).

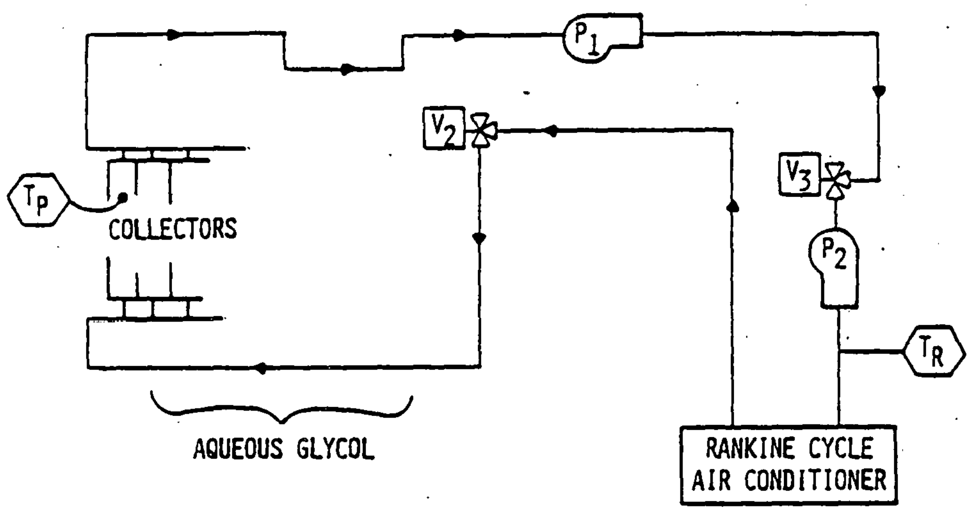



Figure 1-7. Power Generation Schematic

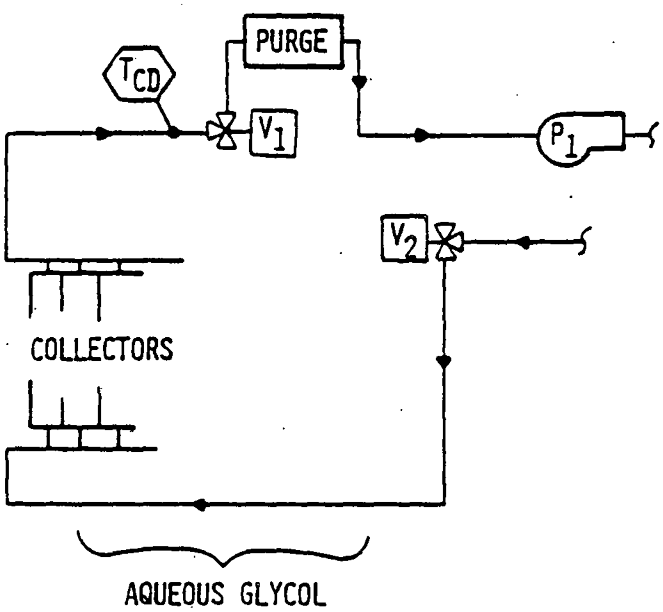



Figure 1-8. Purge Mode Schematic

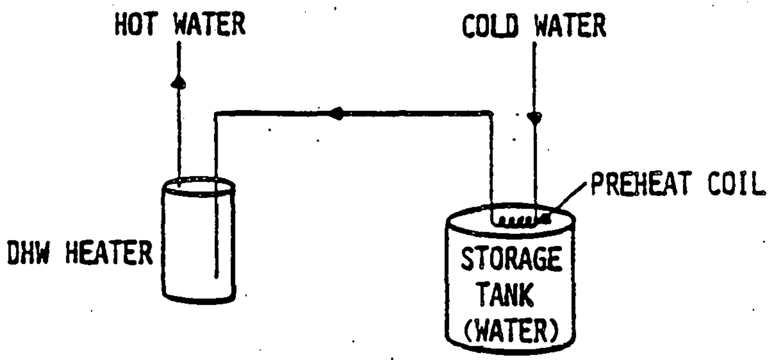



Figure 1-9. Domestic Water Preheating Schematic

## 1.2 TYPICAL SYSTEM OPERATION

The Shenandoah system has two basic operating seasons. During the cooling season the system cools (direct and from storage), preheats domestic hot water, generates electricity, and charges storage. During the heating season the system provides space heating, preheats domestic hot water, and charges storage.

The following sections detail the system operation for a typical day during the heating and cooling seasons.

### 1.2.1 Heating Season Operation

Figure 1-10 shows typical collector array inlet and outlet temperatures during heating season operation (January 13, 1981). During heating season operation the system is activated for energy collection when the difference between the collector absorber plate temperature and the storage tank temperature reaches 18$^{O}$F. Storage charging operation began at 0951 and continued until 1635. Storage charging stops when the difference between the absorber plate temperature and the storage tank temperature falls to 3$^{O}$F. Figure 1-11 shows the available solar insolation during system operation.

Typical storage tank temperatures during heating season operation are shown in Figure 1-12. Fluid is drawn from the top of the storage tank, circulated through a water-to-air heat exchanger in the furnace return air duct, and returned to the middle of the storage tank. There was occasional solar heating throughout the day, with most of the heating being done during the mid-morning and late evening hours. There was a substantial amount of passive heating during the middle of the day. The temperature in the storage tank dropped steadily during the morning until storage charging operation began. The temperature in the storage tank rose steadily until storage charging ended. The upper limit for storage charging during the heating season is 140$^{O}$F.

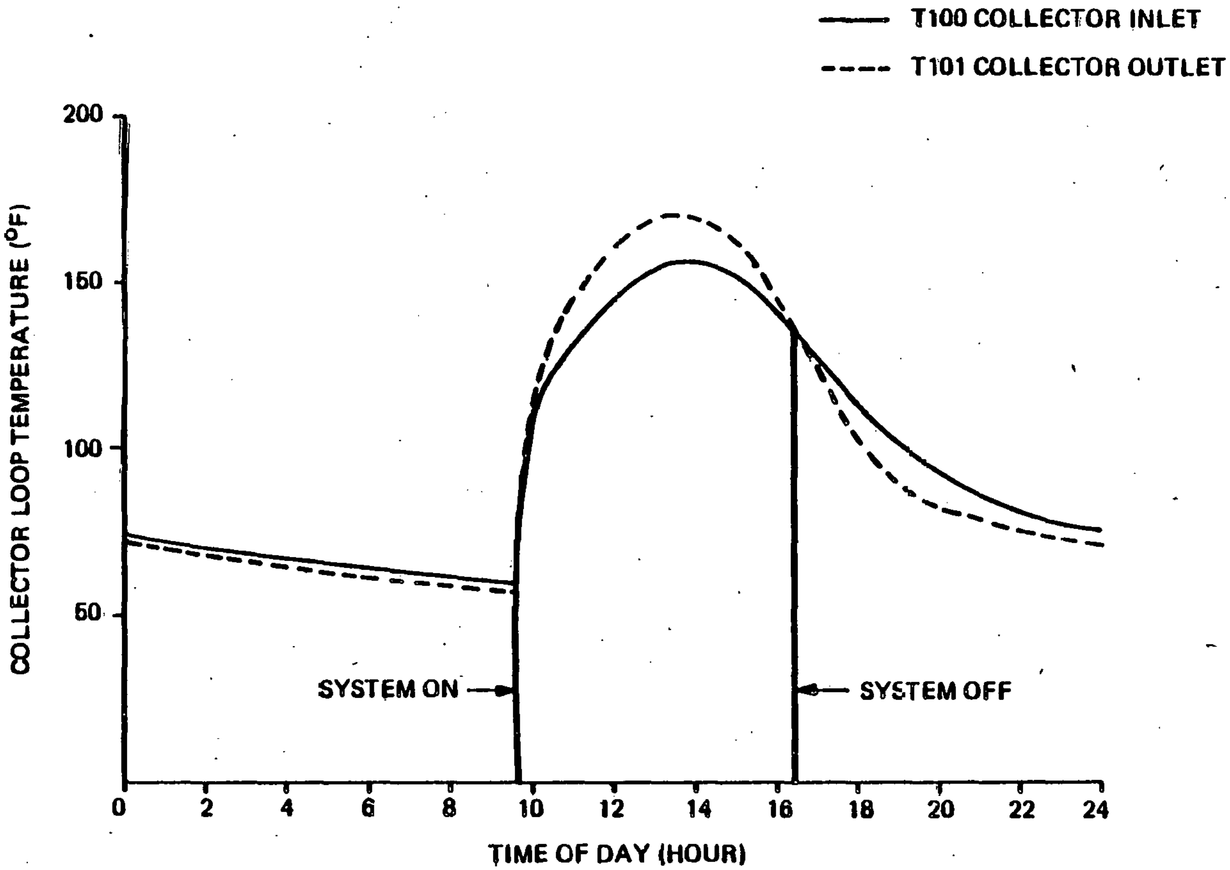



FIGURE 1-10. COLLECTOR ARRAY TEMPERATURES VERSUS TIME OF DAY, HEATING SEASON (01-13-81)

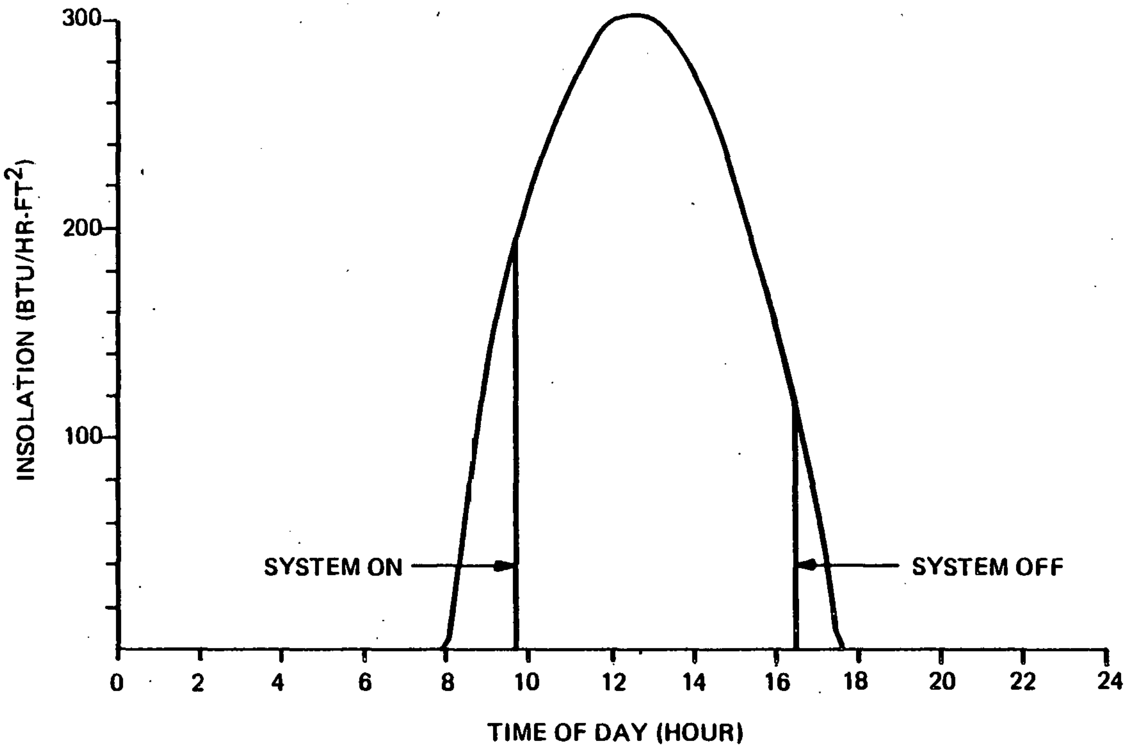



FIGURE 1-11. AVAILABLE SOLAR INSOLATION VERSUS TIME OF DAY, HEATING SEASON (01-13-81)

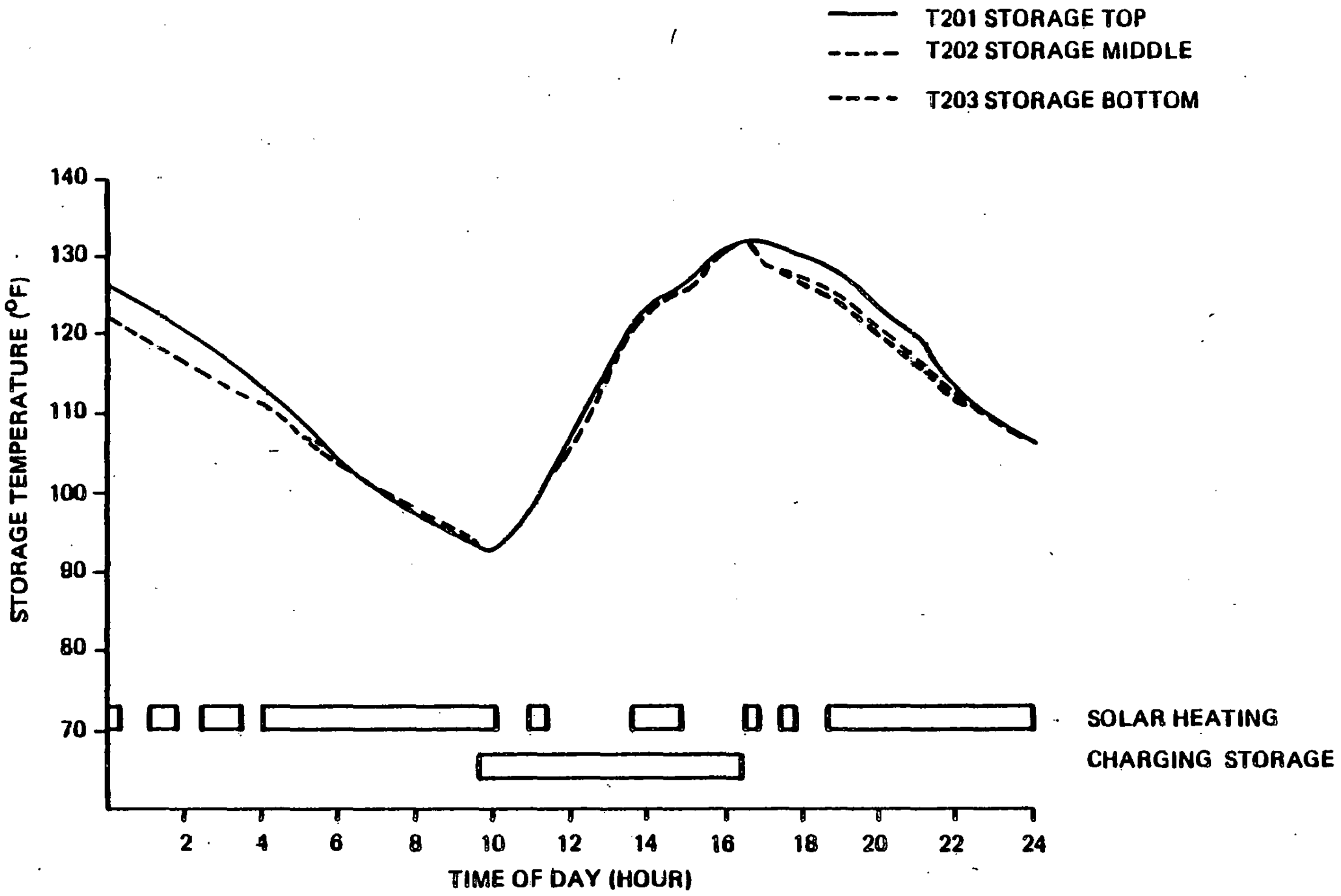



FIGURE 1-12. STORAGE TANK TEMPERATURES VERSUS TIME OF DAY, HEATING SEASON (01-13-81)

### 1.2.2 Cooling Season Operation

Typical collector array inlet and outlet temperatures for cooling season operation are shown in Figure 1-13 for May 8, 1981. At 0853 the collector panel absorber plate temperature reached the operating setpoint of 170°F and fluid was circulated through the collector array. The warm fluid in the array was displaced by the cooler fluid in the system piping. When the cooler fluid reached the absorber plate temperature sensor the flow through the collector array was stopped. After ten minutes the absorber plate temperature again reached 170°F and flow was directed through the collector array. This process continued until all of the fluid in the collector loop was at 170°F. On this day the process was repeated twice. On days when cooling from storage preceded direct cooling or storage charging the fluid will already be warm and the system will go through just one on/off iteration. The drop in the collector array inlet and outlet temperatures at 1221 was due to purge unit activation. The purge unit is activated to reject energy to the environment when the collector array outlet temperature reaches 220°F. The drop in the collector array temperatures at 1248 and rise at 1357 were due to activation and deactivation of the Rankine engine. The available solar insolation is shown in Figure 1-14.

Figure 1-15 shows the storage tank temperatures during cooling season operation. There was no call for cooling in the early morning so the system charged storage from 0920 until 1242. There was direct solar cooling from 1242 until the call for cooling ended at 1352. At 1352 the system began charging storage again. The system continued to charge storage until 1632, when the difference between the collector panel absorber plate temperature and the storage tank temperature fell to 3°F. The collector array outlet temperature, T101, shown in Figure 2.3-4 can be used to approximate the collector panel absorber plate temperature for comparison with the storage tank temperature shown in Figure 2.3-6. At 1730 there was another call for cooling and the system was able to cool from storage until the call for cooling ended at 1914.

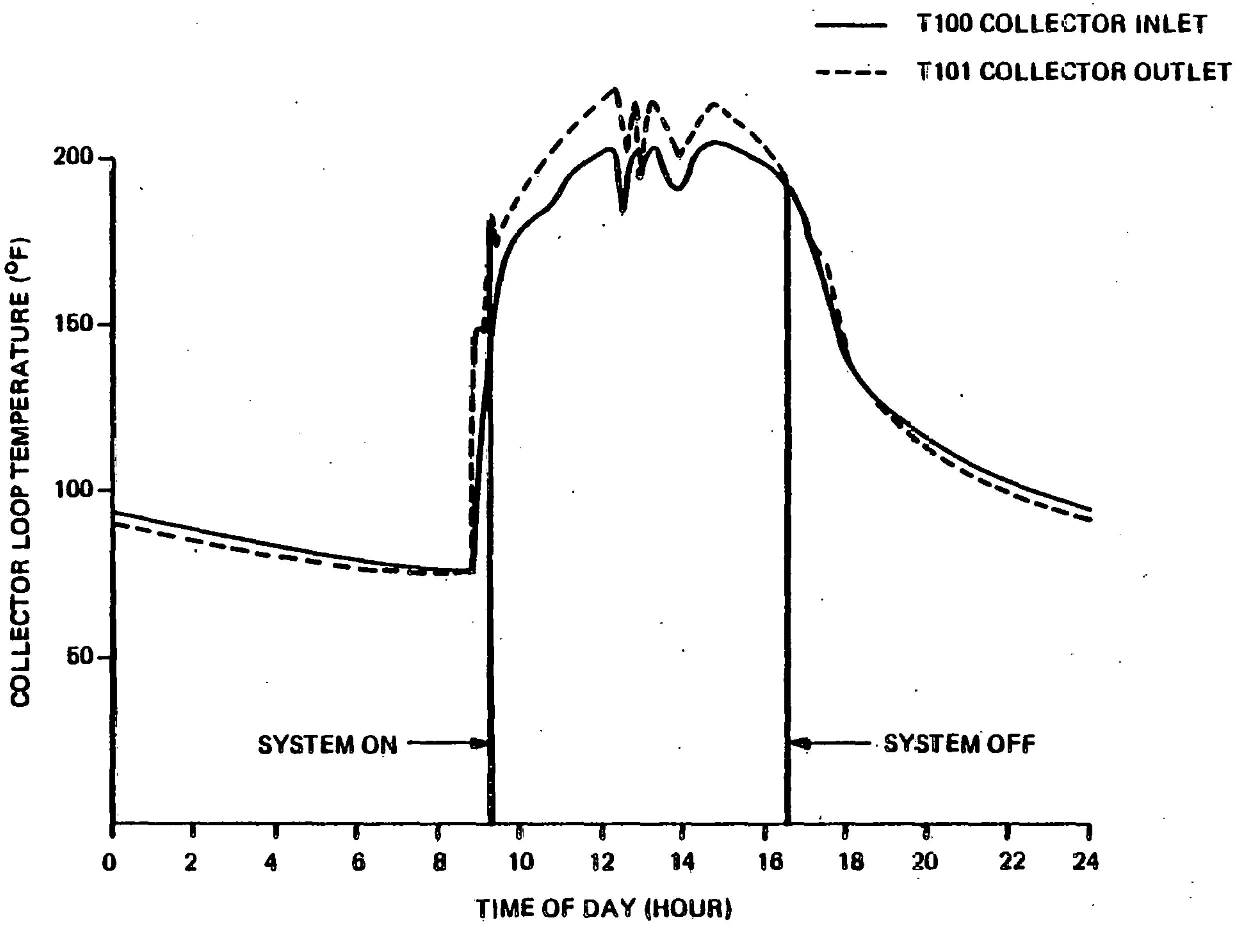



FIGURE 1-13. COLLECTOR ARRAY TEMPERATURES VERSUS TIME OF DAY, COOLING SEASON (05-08-81)

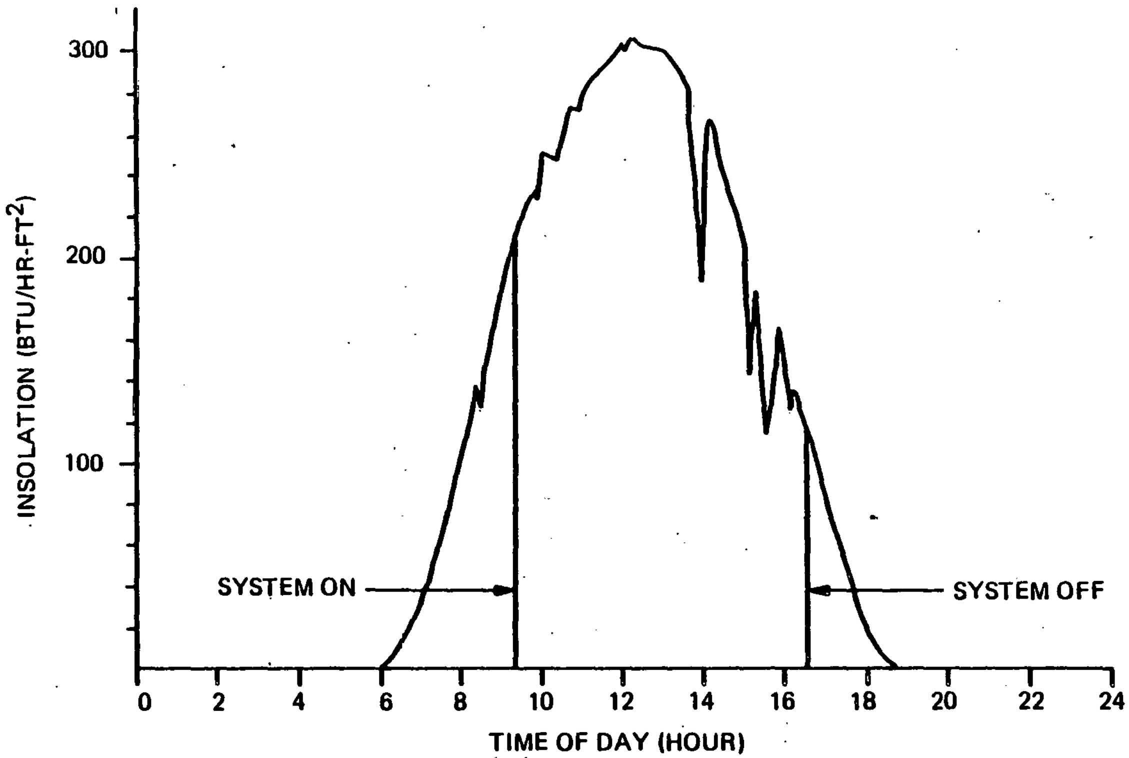



FIGURE 1-14. AVAILABLE INSOLATION VERSUS
TIME OF DAY, COOLING SEASON (05-08-81)

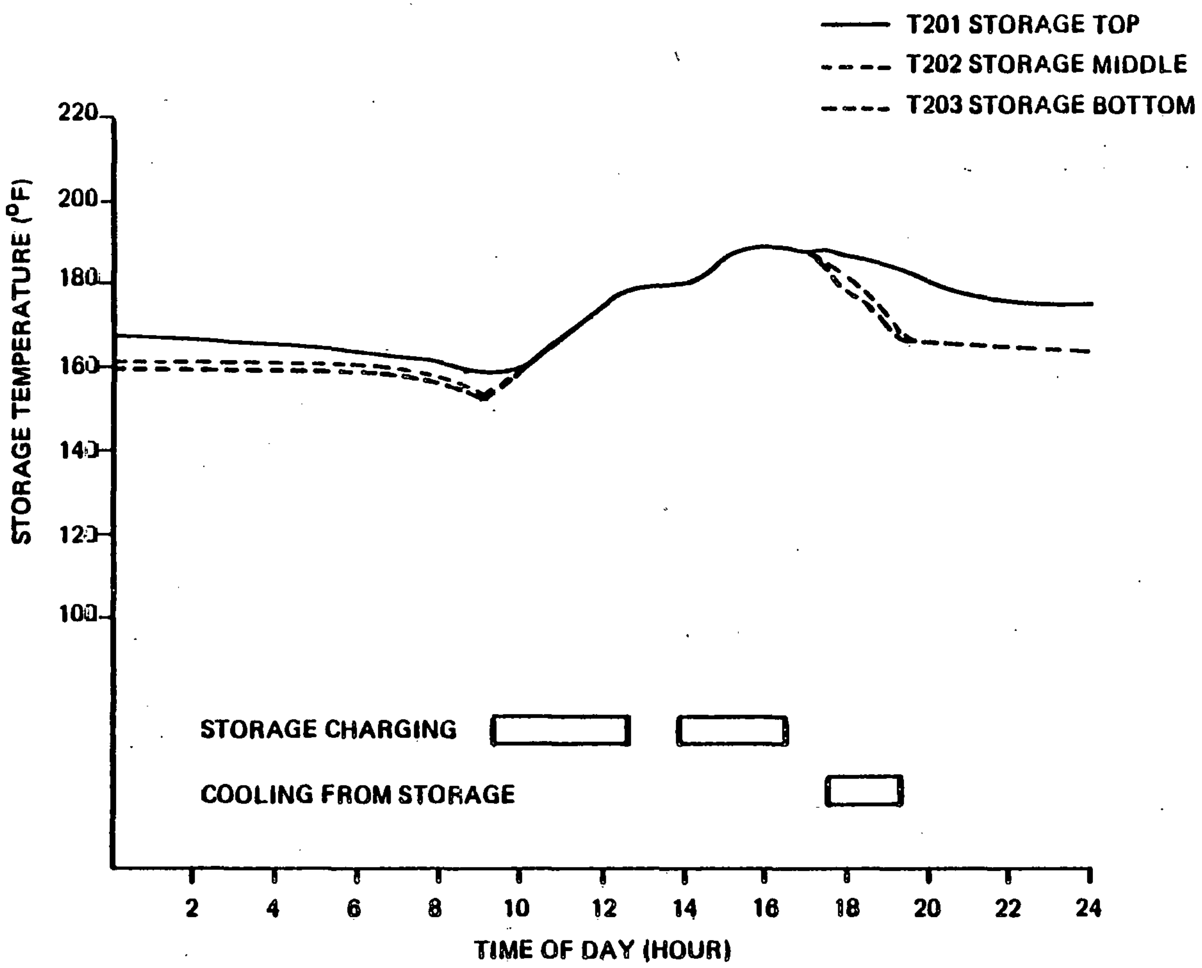



FIGURE 1-15. STORAGE TANK TEMPERATURES VERSUS TIME OF DAY, COOLING SEASON (05-08-81)

The Rankine engine inlet and outlet temperatures are shown in Figure 1-16. There were two periods of Rankine operation during the day. From 1242 until 1352 the Rankine operated direct from the collector array and from 1730 until 1914 the Rankine cooled from storage.

## 1.3 SYSTEM OPERATING SEQUENCE

The system at Shenandoah has two distinct operating seasons as mentioned in Section 1.2. The following sections outline the system operating sequence during the heating and cooling seasons.

### 1.3.1 Heating Season Operating Sequence

The operation of the space heating subsystem is controlled by the building space thermostat. If the stored thermal energy is not able to meet the heating load the auxiliary gas-fired furnace is activated. When there is a sufficient difference between the collector panel absorber plate temperature and the storage tank temperature the collector array is activated for energy collection and storage charging. Domestic hot water is preheated whenever there is hot water usage. Auxiliary hot water heating is available from a gas-fired back-up unit.

The sequence of operation is shown in Figure 1-17 for January 13, 1981. There was solar space heating periodically throughout the day, with the majority of the heating being done in the mid-morning and later in the evening. Auxiliary heating was done almost exclusively in the late evening. The system collected energy and charged storage from 0951 until 1635. A small amount of auxiliary domestic hot water heating was done in the morning to make up for standby losses from the gas-fired water heater.

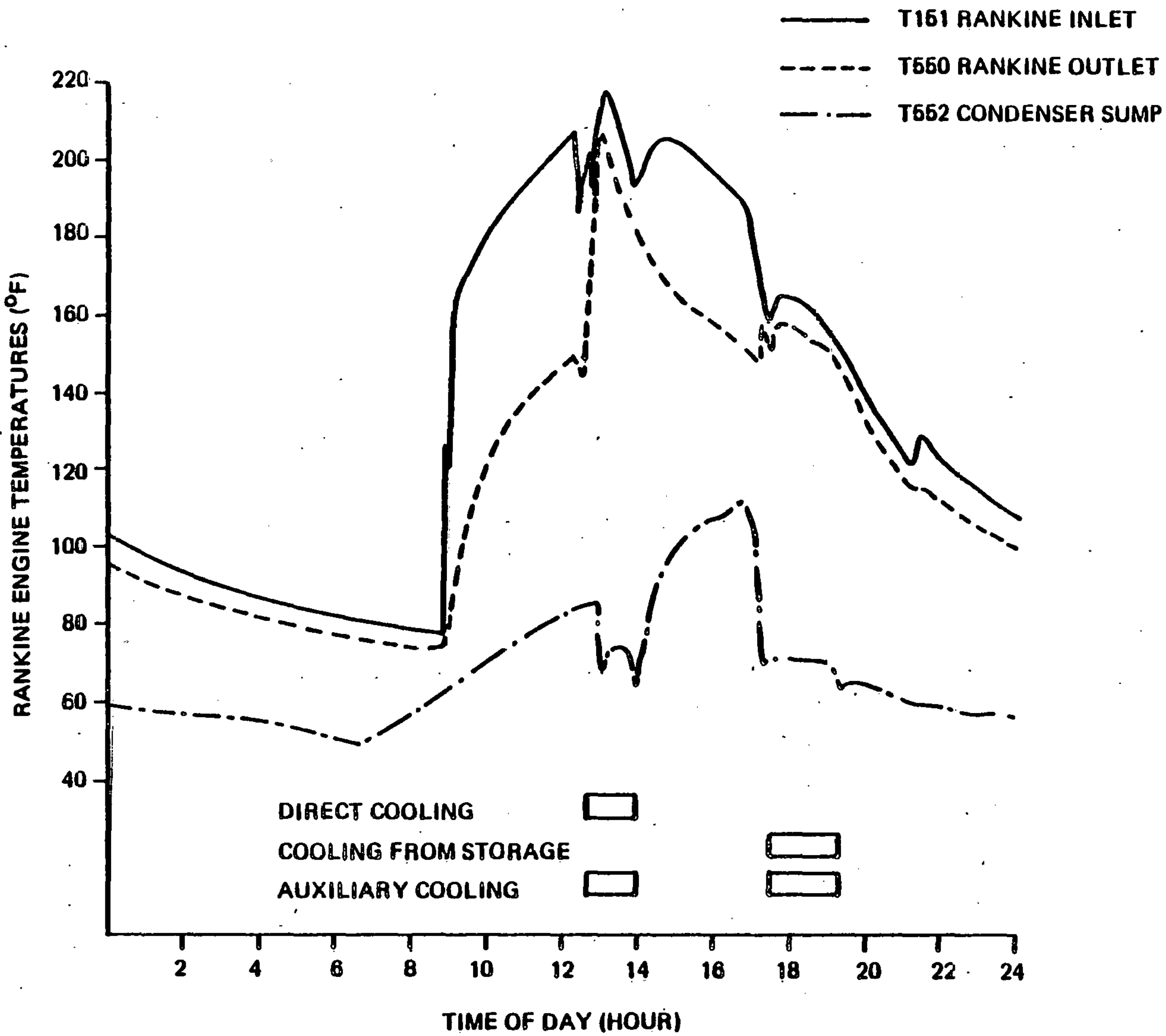



FIGURE 1-16. RANKINE TEMPERATURES VERSUS TIME OF DAY (05-08-81)

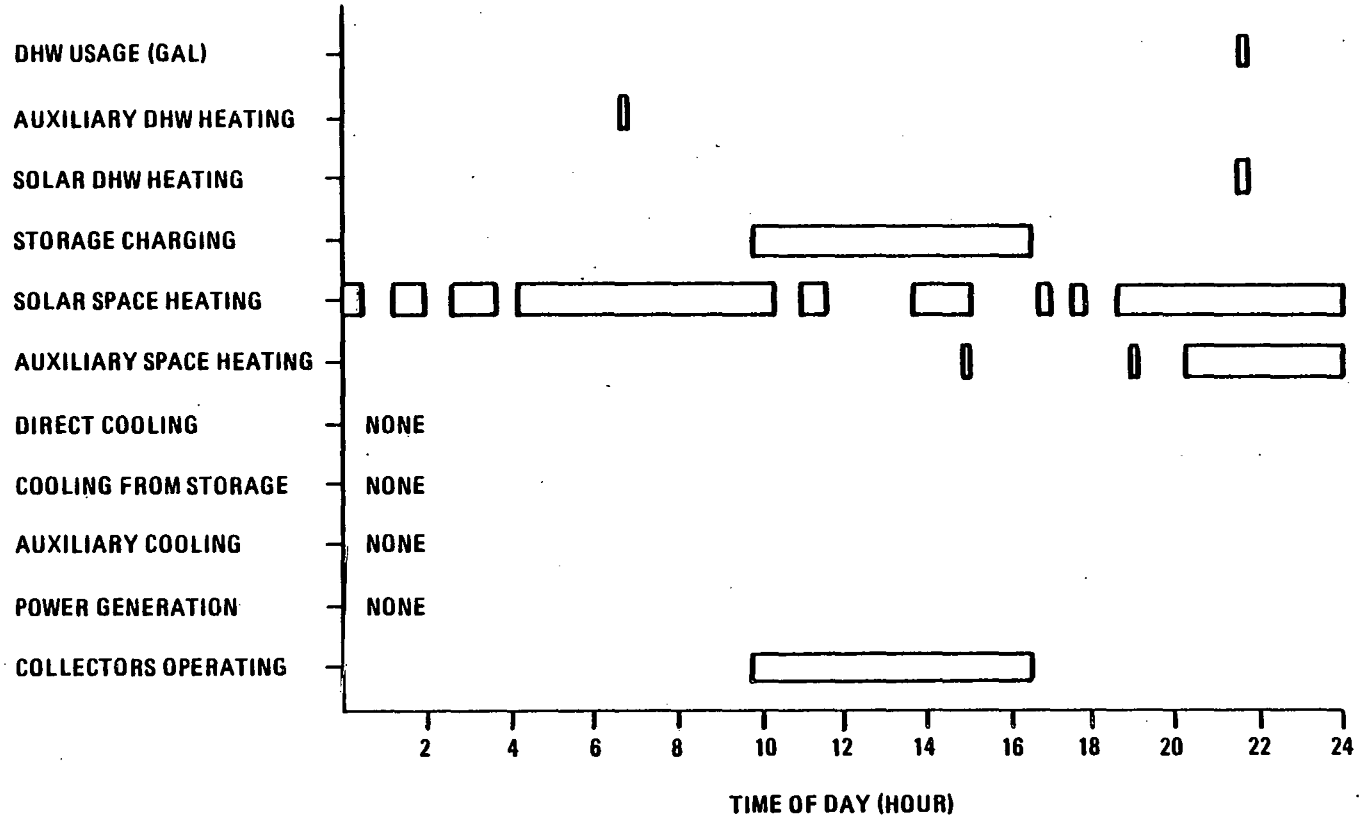



FIGURE 1-17. TYPICAL HEATING SEASON OPERATING SEQUENCE (01/13/81)

### 1.3.2 Cooling Season Operating Sequence

The operation of the space cooling subsystem is controlled by the building space thermostat. Upon a call for cooling, the auxiliary electric motor is activated to drive the compressor. If solar energy is available directly from the collector array the Rankine engine will be activated to unload the auxiliary electric motor. If there is no solar energy available from the collector array and there is sufficient energy in the storage tank the system will operate the Rankine from storage to unload the auxiliary electric motor. If solar energy is not available from either source the compressor will continue to be driven by the auxiliary electric motor. If there is no cooling load the system will charge storage or generate electricity.

The above sequence of operation is shown in Figure 1-18. The collector array was operational from 0920 until 1632. During this time the system charged storage from 0920 until 1242, cooled direct from 1242 until 1352, and charged storage again from 1352 to 1632. Since there was no cooling load in the morning system start-up was determined by the storage tank temperature. Stored thermal energy was used for solar-assisted cooling from 1730 to 1914.

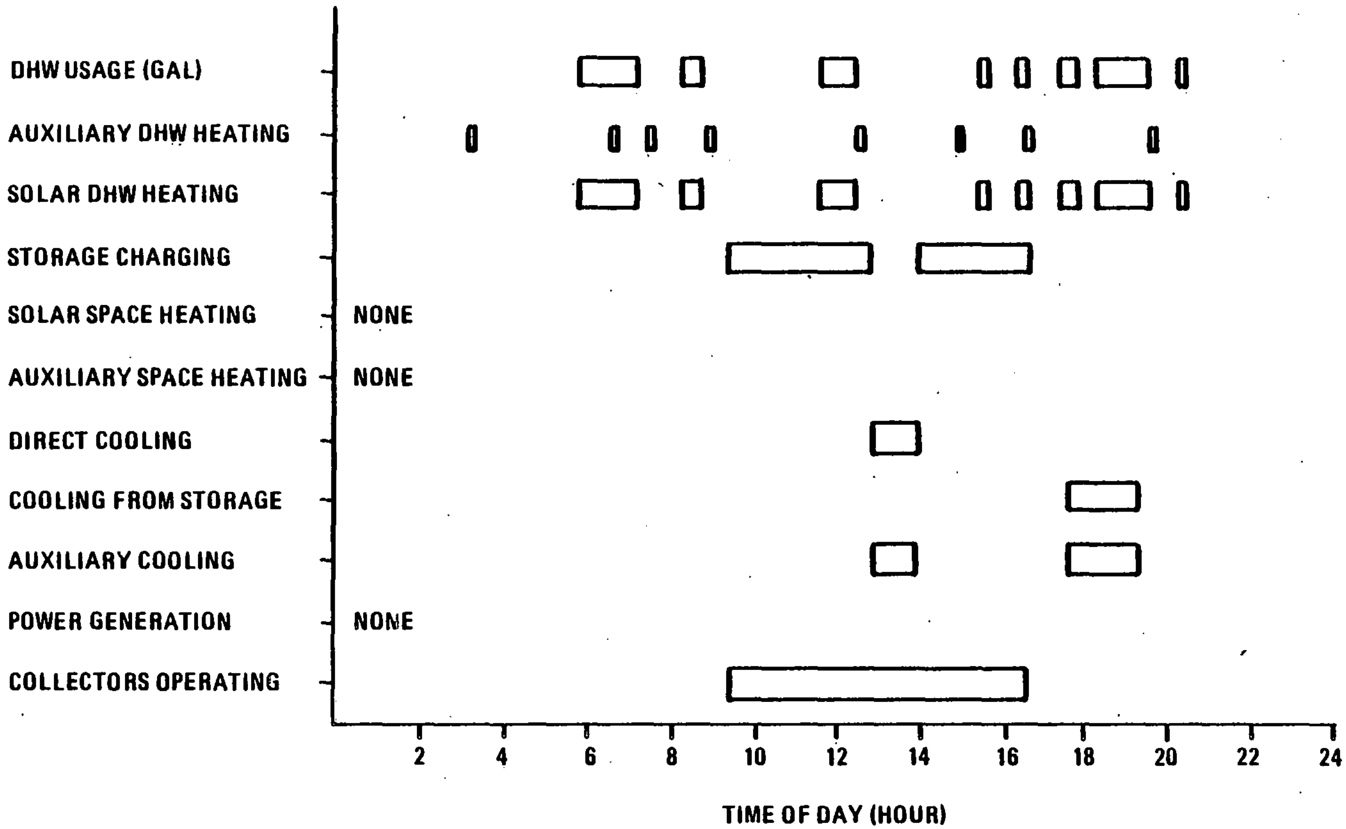



FIGURE 1-18. TYPICAL COOLING SEASON OPERATING SEQUENCE (5/8/81)

# SECTION 2.0
# PERFORMANCE ASSESSMENT

The performance of this test site has been evaluated for The Operational Test Period (OTP) January 1981 through August 1981[1]. Data for this report were gathered and processed through the National Solar Data Program, as described in Appendix C.

The performance assessment for the OTP is made from two perspectives:

- Overall system performance is assessed. The total solar energy available, the system load, and the system solar fraction are presented.
- An in-depth evaluation is made of the performance of the following individual subsystems:

  - Collectors.
  - Storage,
  - Space Heating.
  - Space Cooling.
  - Rankine-cycle air conditioner.

All performance parameters presented in this report conform to the definitions used by the National Solar Data Program for its monthly performance reports [1]; additional parameters have been presented to provide further insight into the performance of this system and its subsystems. The definitions of all performance parameters used in this report are listed in Appendix A. Appendix B contains the equations that were used to calculate the performance parameters from the raw data collected from the site. Appendix C lists the sensors used to monitor the performance of the system; shows the locations within the system of all sensors; and describes the data collection, retrieval and reduction methods.

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(1) Performance during July 1981 is not reported because sufficient measured data was not available either due to data acquisition problems or system problems.

Instrumentation accuracies are affected by sampling error and by systematic sensor errors due to inaccurate calibration, drift and nonlinearities. To evaluate the effect of sensor errors on the performance factors, an error analysis was conducted and is presented in Appendix D. The performance factors presented in this document should be viewed in light of this uncertainty in measurement.

## 2.1 SYSTEM PERFORMANCE

The solar energy system performance summarized in this section can be viewed as the dependent response of the system to certain primary inputs. These primary inputs are incident solar radiation, average ambient temperature and system load. Dependent system responses are the system solar fraction and total energy savings. The monthly values of these inputs and outputs measured during the operational period are shown in Table 2-1, along with long-term average values of daily incident solar energy and outdoor ambient temperature (also see Appendix E). A comparison of measured weather data to the long-term average may be used to indicate expected long-term performance of the system.

Figure 2-1 depicts utilization by the system of the total incident solar energy on the collectors during the OTP. Of the total incident solar radiation (insolation), 69 percent fell on the collectors while the solar pump was active (operational incident energy). The system collected 35 percent of the operational incident solar energy. The collected energy either was delivered to thermal storage to provide for space cooling, space heating, and domestic hot water, delivered to the Rankine-cycle boilers to be used to generate power or to cool the building, purged (intentionally rejected to the environment), or lost to the environment.

TABLE 2-1. SYSTEM PERFORMANCE SUMMARY

| MONTH AND YEAR | AVERAGE DAILY INSOLATION IN THE PLANE OF THE COLLECTOR ARRAY | | AVERAGE AMBIENT TEMPERATURE | | SYSTEM LOAD | | | SOLAR FRACTION OF THE SYSTEM LOAD | | | NET ENERGY SAVINGS | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | MEASURED (Btu\|Ft²-Day) | LONG-TERM AVERAGE (Btu\|Ft²-Day) | MEASURED (°F) | LONG-TERM AVERAGE (°F) | HOT WATER ($10^6$Btu) | SPACE HEATING ($10^6$Btu) | SPACE COOLING ($10^6$Btu) | HOT WATER (PERCENT) | SPACE HEATING (PERCENT) | SPACE COOLING (PERCENT) | FOSSIL FUEL ($10^6$ Btu) | ELECTRICAL ENERGY [kWh(e)] |
| JANUARY 1981 | 1420 | 972 | 39 | 42 | 1.11 | 5.94 | 0 | 25 | 77 | 0 | 6.78 | -73 |
| FEBRUARY 1981 | 1367 | 1205 | 49 | 45 | 1.62 | 2.77 | 0 | 42 | 73 | 0 | 4.08 | -55 |
| MARCH 1981 | 1649 | 1464 | 50 | 51 | 1.57 | 2.40 | 0 | 90 | 95 | 0 | 5.67 | -53 |
| APRIL 1981 | 1559 | 1698 | 67 | 61 | 1.46 | 0.03 | 1.31 | 92 | 0 | 19 | 2.24 | -41 |
| MAY 1981 | 1522 | 1713 | 66 | 69 | 1.56 | 0 | 2.87 | 91 | - | 33 | 2.37 | +44 |
| JUNE 1981 | 1592 | 1704 | 80 | 76 | 1.53 | 0 | 7.96 | 89 | - | 12 | 2.27 | +62 |
| JULY 1981 | (a) | 1643 | (a) | 78 | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) |
| AUGUST 1981 | 1206 | 1656 | 75 | 78 | 1.80 | 0 | 4.71 | 91 | - | 4 | 2.72 | -18 |
| TOTAL | - | - | | | 10.66 | 11.15 | 16.85 | - | - | - | 26.13 | -134 |
| AVERAGE | 1476 | 1507 | 63 | 63 | 1.52 | 3.72 (b) | 4.21 (b) | 76 | 77 | 15 | 3.73 | -19 |

(a) Not available

(b) Seasonal average

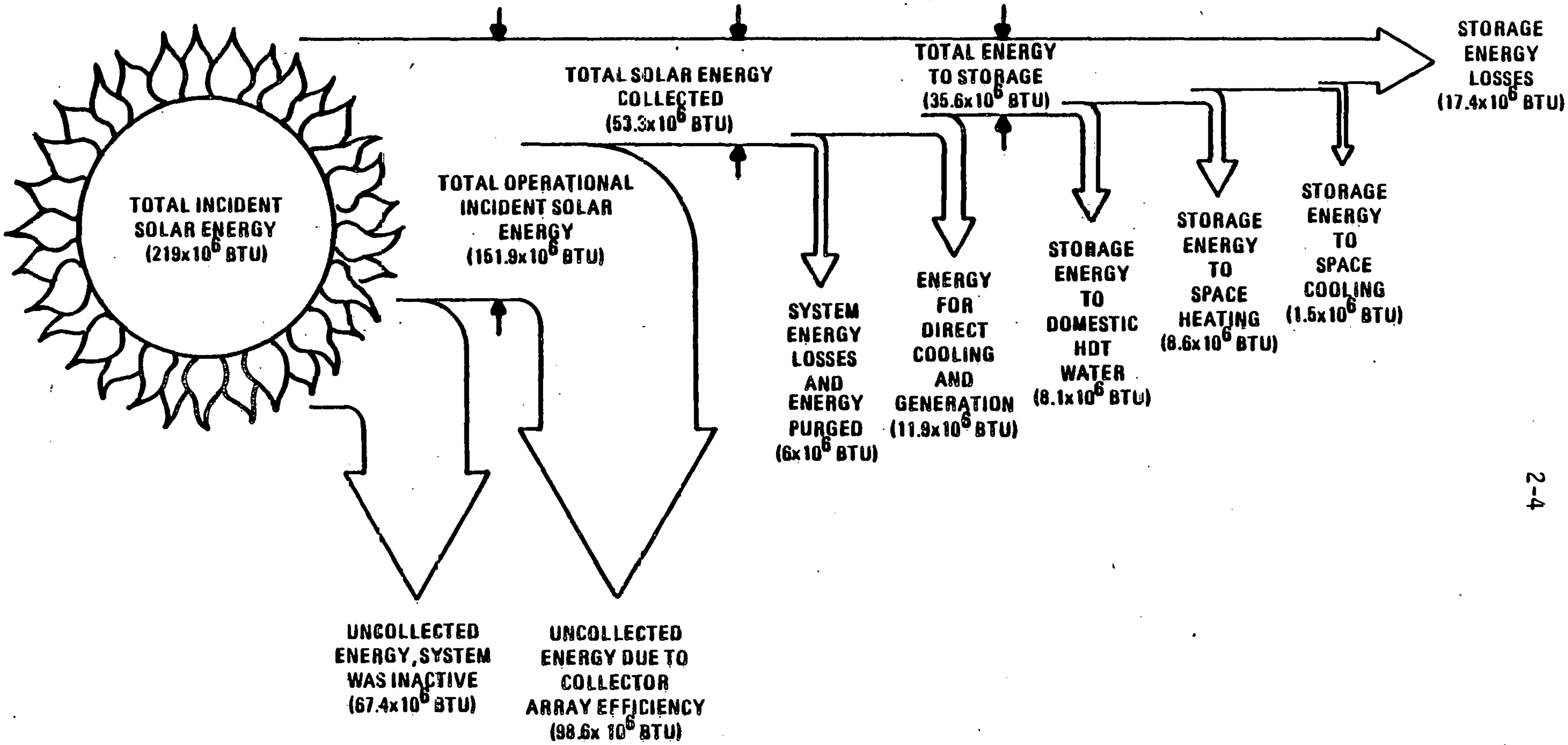



FIGURE 2-1. SYSTEM TEST PERIOD* SOLAR ENERGY USE

* Seven months of OTP (January, February, March, April, May, June, August of 1981)

Of the collected energy 57 percent was used by the load subsystems (22 percent for space cooling, 16 percent for space heating, 15 percent for domestic hot water and 4 percent for power generation). Of the remaining 43 percent of collected energy 42 percent was lost to the environment (32 percent while stored as thermal energy in a thermal storage tank and 12 percent through purging or system losses) and the remaining 1% was left as energy stored at end of OTP. The system energy losses were estimated to be 7% of the energy collected. This includes piping heating losses and the thermal energy required to bring the system to operating temperature.

## 2.2 SUBSYSTEM PERFORMANCE

This subsection presents the results of analyzing the monthly data available for the six subsystems--collector, storage, space heating, space cooling, domestic hot water and Rankine-cycle air conditioner. Subsystem performance is evaluated by calculating a set of primary performance factors. The electrical energy required to operate pumps and fans to support each of these subsystems--while an important consideration in the overall performance of each subsystem--is not presented here but appears in Section 4.0.

### 2.2.1 Collector Performance

The most common measure of collector performance is collector efficiency, defined as the ratio of solar energy collected to total solar energy incident on the array (including the collector frames). Table 2-2 presents the average collector array performance for each month of the reporting period. The collector array efficiency listed in the table was based on total incident solar radiation, including incident solar radiation occurring when the array was not active. Thus, this efficiency was affected directly by system conditions (other than array performance), which determined the active periods of the collector array. The operational collector efficiency, on the other hand, was based only on the solar energy incident upon the array when the array was active (collecting solar energy). This parameter therefore provides a clearer view of the average array performance during operation; it minimizes the effects of other system conditions. The collector array collected 35 percent of

TABLE 2-2. COLLECTOR ARRAY PERFORMANCE

| MONTH AND YEAR | INCIDENT SOLAR ENERGY ($10^6$ Btu) | COLLECTED SOLAR ENERGY ($10^6$ Btu) | COLLECTOR ARRAY EFFICIENCY (Percent) | OPERATIONAL INCIDENT SOLAR ENERGY ($10^6$ Btu) | OPERATIONAL COLLECTOR ARRAY EFFICIENCY (Percent) | DAYTIME AMBIENT TEMPERATURE (°F) |
|---|---|---|---|---|---|---|
| JANUARY 1981 | 30.8 | 7.1 (a) | 23 | 21.0 | 34 | 46 |
| FEBRUARY 1981 | 26.9 | 6.3 (a) | 24 | 20.4 | 31 | 54 |
| MARCH 1981 | 35.9 | 8.8 | 25 | 26.2 | 34 | 58 |
| APRIL 1981 | 32.8 | 8.1 | 25 | 22.3 | 36 | 76 |
| MAY 1981 | 33.1 | 8.3 | 25 | 23.2 | 36 | 75 |
| JUNE 1981 | 33.5 | 8.9 | 27 | 22.1 | 40 | 88 |
| JULY 1981 | (b) | (b) | (b) | (b) | (b) | (b) |
| AUGUST 1981 | 26.3 | 5.8 | 22 | 16.7 | 35 | 83 |
| TOTAL | 219.3 | 53.3 | -- | 151.9 | -- | -- |
| AVERAGE | 31.3 | 7.6 | 24 | 21.7 | 35 | 69 |

(a) Estimated using available data. Collector outlet temperature sensor was inaccurate.

(b) Not available.

the solar energy incident during collector operation during the Operational Test Period.

The array did better based on an instantaneous basis. Figure 2-2 depicts the instantaneous collector efficiency vs. the operating point. The operating point is defined as:

$$x_j = \frac{T_i - T_a}{I}$$

where,

$X_j$ = collector operating point at the jth instant,
$T_i$ = collector inlet temperature,
$T_a$ = ambient temperature, and
$I$ = insolation.

All points for calculation of the measured collector efficiencies were taken within ± 1 hour of noon and when the solar insolation was steady. The figure also illustrates the measured and expected array performance. The performance curves are all based on gross collector area and include:

- The performance of a single collector panel before and after a long-term weathering test consisting of 15-1/2 months of stagnation and exposure to the environment [2].

- Measured collector array data, calculated as the energy gain of fluid passing through the collectors divided by solar insolation striking the gross collector area during system operation. All points for calculation of measured collector efficiency were taken within ± 2 hours of noon.

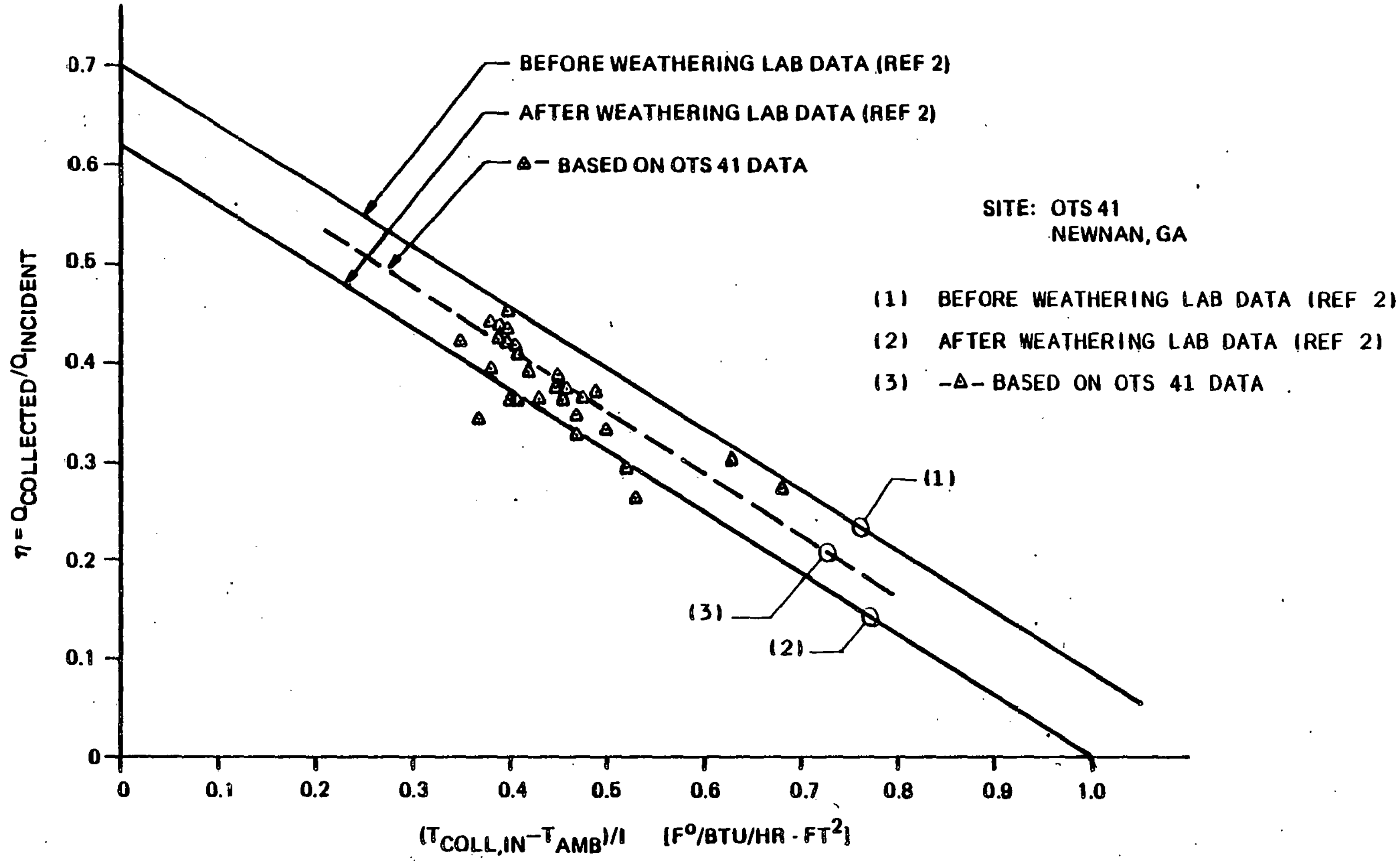



FIGURE 2-2. COLLECTOR EFFICIENCY VERSUS OPERATING POINT FOR SHENANDOAH COLLECTOR ARRAY.

As shown, the measured performance of the array was better than a single panel after prolonged stagnation conditions but was poorer than the before weathering single panel performance. The performance of the array normally would be expected to be worse than that of a single panel because of heat losses from the array piping and because of the effects of connecting several collectors in series. The array did remain in a stagnation condition for several days during system installation and during test period, but the stagnation did not degrade the array performance below that of a single panel that was stagnated for 15-1/2 months.

### 2.2.2 Storage Subsystem

The storage subsystem consists of an epoxy-lined 1000-gallon capacity steel storage tank filled with approximately 875 gallons of water containing a corrosion inhibitor. The tank is insulated with a 10-inch layer of fiberglass insulation. All piping into and out of the storage tank is insulated with 3/4-inch Armaflex foam insulation. The tank is maintained at atmospheric pressure through an atmospheric vent. It is located in the mechanical room in the basement of the house. It interfaces with the collector loop via a shell and tube heat exchanger.

Table 2-3 lists the storage subsystem performance parameters for each month of the test period. Energy to storage is the total solar energy delivered to the storage subsystem by the collector subsystem. Energy from storage is the total solar energy transferred from the storage subsystem to the load subsystems. Storage efficiency is a measure of the portion of the energy delivered to storage that was delivered to the load subsystems or resulted in a change in stored energy. It was calculated as the ratio of the sum of the energy from storage and the change in stored energy to the energy to storage. Complete definitions of the performance factors listed in Table 2-3 are given in Appendix A.

TABLE 2-3. STORAGE SUBSYSTEM PERFORMANCE

| MONTH | ENERGY TO STORAGE ($10^6$ Btu) | ENERGY FROM STORAGE ($10^6$ Btu) | CHANGE IN STORED ENERGY ($10^6$ Btu) | STORAGE HEAT LOSS ($10^6$ Btu) | STORAGE TEMPERATURE | | | EFFECTIVE STORAGE HEAT LOSS COEFFICIENT (Btu/hr-ft$^2$-F) | STORAGE EFFICIENCY (PERCENT) |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | AVERAGE (°F) | DAY 1 (°F) | DAY 30[a] (°F) | | |
| JANUARY 1981 | 6.4 | 4.6 | -0.02 | 1.8 | 125 | 127 | 125 | 0.30 | 72 |
| FEBRUARY 1981 | 5.4 | 2.7 | +0.55 | 2.1 | 141 | 125 | 195 | 0.31 | 60 |
| MARCH 1981 | 7.0 | 3.7 | -0.12 | 3.4 | 176 | 194 | 179 | 0.31 | 51 |
| APRIL 1981 | 4.6 | 1.5 | +0.14 | 3.0 | 174 | 179 | 197 | 0.30 | 36 |
| MAY 1981 | 4.5 | 2.3 | -0.40 | 2.6 | 171 | 197 | 146 | 0.24 | 42 |
| JUNE 1981 | 3.5 | 1.6 | +0.27 | 1.6 | 150 | 146 | 130 | 0.21 | 53 |
| JULY 1981 | (b) | (b) | (b) | (b) | (b) | (b) | (b) | (b) | (b) |
| AUGUST 1981 | 4.2 | 2.0 | +0.02 | 2.2 | 175 | 172 | 175 | 0.21 | 48 |
| TOTAL | 35.6 | 18.4 | 0.44 | 16.7 | -- | 127 | 175 | -- | -- |
| AVERAGE | 5.1 | 2.63 | 0.06 | 2.4 | 160 | -- | -- | 0.26 | 53 |

(a) Last day of the month
(b) Not available

The performance of the storage subsystem during the heating season (December through March) was typically better than it was during the rest of the test period, as based on storage efficiency. Additionally, the storage efficiency was generally better during periods of high system load (heating season) than it was during periods of low system load (spring and fall). The periods of lowest storage efficiency were typified by high storage temperatures and high storage heat losses. The storage temperature was necessarily kept high during the cooling season (May through August) because a minimum storage temperature of $160^{o}$F was needed to cool from storage. During the month of March, the storage temperature remained high because the Rankine-cycle engine was not available to generate electricity and the system thus delivered as much collected solar energy as possible to the storage subsystem. During the spring or fall, when the Rankine engine is operational (such as April 1981), the system will charge the storage tank to $160^{o}$ or $200^{o}$F, depending on whether the house thermostat is set in the "heat" mode or the "cool" mode, respectively. Thereafter, any additional collected solar energy will be used to generate electricity (see Subsection 2.2.6). During April, the thermostat mode setting was "cool" for most of the month, resulting in a high average storage temperature.

The storage heat loss includes the heat losses from the storage tank and the piping to and from the storage tank and the sensor locations on the piping. The heat loss from piping was estimated to be 6% of the total heat loss. The storage heat loss coefficient based on bare tank surface area was calculated to be 0.26 Btu/hr-$ft^2$-F, while the heat loss coefficient based on outside tank area is 0.14 Btu/hr-$ft^2$-F. This is larger than anticipated. A vertical cylindrical tank, 5.5 feet in diameter and 5.5 feet high, insulated with 10 inches of fiberglass on the top and sides and 5 inches on the bottom, should have an equivalent heat loss coefficient of 0.05 Btu/hr-$ft^2$F.

### 2.2.3 Hot Water Subsystem

The domestic hot water (DHW) subsystem at Shenandoah consists of a finned-tube coil suspended in the top of the storage tank and a 52-gallon conventional gas domestic hot water heater. Water is preheated as it passes through the finned-tube coil and then enters the DHW heater, from which it is drawn upon demand into the house. The DHW heater serves to store hot water and to provide any necessary auxiliary heating. Auxiliary heating is required to raise the preheated water to the desired delivery temperature (approximately 140°F), if necessary, and to maintain the hot water in the heater at the delivery temperature. Since hot water heaters often are not well-insulated, thermal losses from the water in the heater may be large and must be made up by auxiliary heating during periods of low hot water usage (and hence low preheating).

The performance of the DHW subsystem is summarized in Table 2-4. The hot water usage at the house was very low the first half of January 1981 due to nonoccupancy of the house. The DHW solar fraction was low throughout the month of January and part of February due to several factors:

- The average storage tank temperature was fairly low resulting in low preheating.
- The DHW heater was turned on high, keeping the water at 160 °F to 170°F, resulting in large DHW tank thermal losses.
- The largest hot water demands occurred on days when the storage tank was not very hot.

In the middle of February the DHW temperature limit at the gas-fired water heater was lowered and on February 21, 1981 the storage charging setpoint was raised from 140°F to 190°F. At the previous 140°F heating setpoint the storage temperature was less than the desired DHW temperature and it was not possible to meet the DHW load with solar input alone. The solar DHW system

TABLE 2-4. HOT WATER SUBSYSTEM PERFORMANCE

| MONTH | ENERGY SUPPLIED | | | | HOT WATER DEMAND | | AVERAGE DAILY USAGE | HOT WATER STANDBY LOSSES | SUPPLY WATER TEMPERATURE | WEIGHTED SOLAR FRACTION |
|---|---|---|---|---|---|---|---|---|---|---|
| | AUXILIARY ($10^6$ Btu) | AUXILIARY THERMAL ($10^6$ Btu) | SOLAR ($10^6$ Btu) | TOTAL ($10^6$ Btu) | ($10^6$ Btu) | (Gal.) | (Gal.) | ($10^6$ Btu) | ($^oF$) | (Percent) |
| JANUARY | 1.393 | 0.836 | 0.279 | 1.115 | 0.763 | 863 | 27 | 0.352 | 55 | 25 |
| FEBRUARY | 1.573 | 0.944 | 0.680 | 1.624 | 1.169 | 1349 | 48 | 0.455 | 56 | 42 |
| MARCH | 0.265 | 0.159 | 1.407 | 1.566 | 1.271 | 1793 | 58 | 0.295 | 58 | 90 |
| APRIL | 0.190 | 0.114 | 1.345 | 1.459 | 1.158 | 1757 | 59 | 0.301 | 65 | 92 |
| MAY | 0.237 | 0.142 | 1.422 | 1.564 | 1.242 | 1859 | 60 | 0.322 | 66 | 91 |
| JUNE | 0.287 | 0.172 | 1.358(a) | 1.530(a) | 1.224(a) | 1800(a) | 60(a) | 0.306(a) | 72 | 89(a) |
| JULY | (b) | (b) | (b) | (b) | (b) | (b) | (b) | (b) | (b) | (b) |
| AUGUST | 0.287 | 0.172 | 1.633 | 1.805 | 1.467 | 2327 | 75 | 0.338 | 70 | 91 |
| TOTAL | 4.232 | 2.539 | 8.124 | 10.660 | 8.294 | 11748 | -- | 2.369 | -- | -- |
| AVERAGE | 0.605 | 0.363 | 1.161 | 1.523 | 1.185 | 1678 | 56 | 0.338 | -- | 76 |

(a) Estimated

(b) Not available

performance for the reamining months of OTP had a solar contribution of about 90%. The performance for June 1981 was estimated using data for previous three months. This was because of a faulty DHW flowmeter data was not available.

The total thermal energy supplied to the DHW subsystem was greater than the hot water demand and was equal to the sum of the hot water demand and the standby losses. The standby losses were about 19 percent of the total thermal energy supplied to the tank. These losses were a function primarily of the difference between the hot water temperature and the surrounding air temperature and thus were fairly constant. The solar fraction was calculated by apportioning the standby losses between solar and auxiliary heating according to how much thermal energy was supplied by each. The solar contribution for the OTP was calculated to be 76 percent. Also notice, the auxiliary thermal energy is calculated by multiplying auxiliary energy (heating value of natural gas used) by the furnace efficiency of 0.6.

Because of inherent losses associated with the thermal storage subsystem, solar thermal energy in excess of that used by the DHW subsystem should be collected and stored. To provide for 8.12 MMBtu of thermal energy from thermal storage for DHW, about 17.3 MMBtu of solar thermal energy were collected and stored. This is calculated by utilizing the storage efficiencies for each month presented in table 2-3. For example for the month of January, solar energy collected to provide for DHW is calculated by dividing the energy from storage (0.279 MMBtu) by storage efficiency from table 2.3 (0.716). This is calculated to be 0.390 MMBtu.

### 2.2.4 Space Heating Subsystem

The Shenandoah space heating subsystem consists of a conventional gas-fired forced-air furnace with a water-to-air heat exchanger located in the return air duct. Hot water is supplied to the solar heating coil directly from the storage tank. The performance parameters for the space heating subsystem for the entire test period are listed in Table 2-5. The solar system provided 77 percent of the heating load of $11.2 \times 10^6$ Btu during the heating season.

Because of the inherent losses associated with the thermal storage subsystem, thermal energy in excess of that required for space heating should be collected and stored. To provide 8.63 MMBtu of thermal energy for space heating from thermal storage, about 13.9 MMBtu of solar energy was required to be collected and stored. This is readily calculated by utilizing the storage efficiencies during the heating season presented in table 2-3.

When the storage tank was hot, solar heating supported most or all of the heating load. If the storage tank was above 100°F no auxiliary heating was needed. In fact, on the coldest day of the heating season, January 13, 1981, solar heating supplied 100% of the load during the coldest part of the day, 4:00 a.m. until 10:00 a.m. The average ambient temperature during this period was 14°F, with 3 hours of temperatures of 10°F or colder. The house temperature during the period was 63°F. The solar system ran continuously in the solar heating mode for the entire period. The storage tank temperature fell from 107°F to 93°F over the same period. The average heating load for the period was 20,000 Btu/hr. The operating energy for continuous operation such as this is significant. The solar heating pump consumes 170 watts of electricity while the furnace fan requires 205 watts. The resulting natural gas savings are high, however. The operating energy and energy savings for heating subsystem are presented in Sections 3.0 and 4.0, respectively.

TABLE 2-5. SPACE HEATING SUBSYSTEM PERFORMANCE

| MONTH | SPACE HEATING LOAD ($10^6$ Btu) | ENERGY SUPPLIED | | | SOLAR FRACTION (Percent) | BUILDING DRY BULB TEMPERATURE (°F) | AMBIENT TEMPERATURE (°F) |
|---|---|---|---|---|---|---|---|
| | | AUXILIARY FUEL ($10^6$ Btu) | AUXILIARY THERMAL ($10^6$ Btu) | SOLAR ($10^6$ Btu) | | | |
| JANUARY 1981 | 5.94 | 2.34 | 1.61 | 4.33 | 77 | 69 | 39 |
| FEBRUARY 1981 | 2.776 | 1.10 | 0.76 | 2.02 | 73 | 72 | 49 |
| MARCH 1981 | 2.40 | 0.18 | 0.12 | 2.28 | 95 | 71 | 50 |
| APRIL 1981 | 0.03 | 0.05 | 0.03 | 0 | 0 | 75 | 67 |
| MAY 1981 | 0 | 0 | 0 | 0 | -- | 75 | 80 |
| JUNE 1981 | 0 | 0 | 0 | 0 | -- | 75 | 80 |
| JULY 1981 | (a) | (a) | (a) | (a) | (a) | (a) | (a) |
| AUGUST 1981 | 0 | 0 | 0 | 0 | -- | 74 | 75 |
| TOTAL | 11.15 | 3.67 | 2.52 | 8.63 | -- | -- | -- |
| AVERAGE[(b)] | 3.72 | 1.22 | 0.84 | 2.88 | 77 | 74 | -- |

(a) Not available

(b) Seasonal average based on January through March data.

The solar energy to heating values presented in Table 2-5 do not include any portion of the storage heat losses, which did serve indirectly to reduce the heating load in the winter by raising the basement air temperature and reducing heat losses from the conditioned space. The storage heat losses did not, however, contribute directly to the heating of the conditioned space. The storage losses would, of course, serve to increase the heating solar fraction if they were included in the solar energy to heating.

### 2.2.5 Space Cooling Subsystem

The Shenandoah space cooling subsystem is an organic Rankine-cycle engine and electric-motor-driven air conditioner with a direct expansion evaporator coil located in the supply air duct inside the house. All components of the subsystem except the evaporator coil are located in a single outdoor unit located behind the garage. The furnace fan is used to deliver cool air to the house.

Measured monthly performance parameters for the space cooling subsystem are listed in table 2-6 for the test period. Significant cooling load occurred during June 1981. The circulating fan was operational for over 60 percent of the time during the month. In June, 27 percent of the cooling load was met while the Rankine engine was actually operating. Over the entire cooling season, during the OTP, 30 percent of the cooling load was met while the Rankine engine was operating. Of this 83 percent was during direct cooling and the remainder during cooling from storage. Since the Rankine engine was not designed to produce enough shaft power to provide all the power needed by the air-conditioner, some auxiliary electric was always needed to meet the cooling load. Because of this, the solar cooling fraction was 15 percent.

TABLE 2-6 SPACE COOLING SUBSYSTEM PERFORMANCE

| MONTH | COOLING LOAD | | | | SOLAR ENERGY USED | | AUXILIARY ELECTRIC FUEL | | | AMBIENT TEMPER-ATURE | DAYTIME AMBIENT TEMPER-ATURE | BUILDING DRY BULB TEMPER-ATURE |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | TOTAL | DURING DIRECT COOLING | DURING COOLING FROM STORAGE | SOLAR FRACTION OF LOAD | DIRECT FROM COLLECTORS | FROM STORAGE | TOTAL | DURING SOLAR COOLING (DIRECT) | DURING STORAGE COOLING | | | |
| | ($10^6$ Btu) | ($10^6$ Btu) | ($10^6$ Btu) | (PERCENT) | ($10^6$ Btu) | ($10^6$ Btu) | [$10^6$ Btu(e)] | [$10^6$ Btu(e)] | [$10^6$ Btu(e)] | (°F) | (°F) | (°F) |
| January 1981 | 0 | 0 | 0 | - | 0 | 0 | 0 | 0 | 0 | 39 | 46 | 69 |
| February 1981 | 0 | 0 | 0 | - | 0 | 0 | 0 | 0 | 0 | 49 | 54 | 72 |
| March 1981 | 0 | 0 | 0 | - | 0 | 0 | 0 | 0 | 0 | 50 | 58 | 71 |
| April 1981 | 1.31 | 0.46 | 0 | 19 | 0.93 | 0 | 0.38 | 0.10 | 0 | 67 | 76 | 75 |
| May 1981 | 2.87 | 1.36 | 0.66 | 33 | 2.91 | 0.87 | 0.75 | 0.31 | 0.14 | 66 | 75 | 74 |
| June 1981 | 7.96 | 2.07 | 0.06 | 12 | 4.88 | 0.26 | 2.94 | 0.63 | 0.05 | 80 | 88 | 75 |
| July 1981 | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) |
| August 1981 | 4.71 | 0.32 | 0.14 | 6 | 1.22 | 0.37 | 2.59 | 0.17 | 0.06 | 75 | 83 | 74 |
| TOTAL | 16.85 | 4.21 | 0.86 | - | 9.94 | 1.50 | 6.66 | 1.21 | 0.25 | - | - | - |
| AVERAGE | 4.21 | 10.5 | 0.22 | 15 | 2.49 | 0.38 | 1.67 | 0.30 | 0.06 | - | - | - |

(a) Not available

The solar cooling fraction was calculated as the ratio of the total turbine work output (cooling only) to the sum of the turbine output and the motor output. The motor shaft output was estimated as the product of the electrical energy used and the motor efficiency. The turbine output was calculated as the product of the Rankine-cycle thermal efficiency and the solar energy used by the Rankine engine. The thermal efficiency was estimated from a lookup table as a function of solar inlet temperature and condenser sump temperature (see Subsection 2.2.6). However, the Rankine engine efficiencv was lower than expected, during most of OTP, as will be discussed in subsequent section. The actual solar contribution was lower than reported in table 2-6.

Also notice from table 2-6 the ratio of auxiliary cooling load to auxiliary energy used in auxiliary mode for the season is 2.3. For June this ratio is 2.6. Under design conditions this ratio should have been 3.7. This lower performance was due to the fact that the air-conditioner did not produce 2.9 tons of cooling as designed.

### 2.2.6 Rankine-Cycle Engine

At Shenandoah the solar energy used for space cooling is converted to shaft work (which is used to run a conventional vapor compression refrigeration cycle air conditioner) by an organic Rankine-cycle engine. The hot solar fluid is used to boil the Rankine-cycle working fluid (Freon 113), which is used to run a high-speed turbine. The high-speed turbine is connected to the low-speed (1750 RPM) motor through a single-reduction gearbox. The gearbox is coupled to one end of the motor shaft through an over-running clutch, which allows the motor to run independently of the Rankine engine when it is not operating, but which also allows the Rankine engine to assist the electric motor in driving its load. The opposite end of the motor shaft is coupled through an electric clutch to the compressor for the air conditioner. In this configuration, the motor may run the compressor by itself or with assistance from the Rankine-cycle engine. All components of the Rankine-cycle-air conditioner (RC/AC) except for the air conditioner's evaporator coil are located in a

single outdoor unit. The unit contains the RC/AC controls, an evaporative condenser for each section (Rankine engine and air conditioner), and the entire RC/AC drive train, (Rankine engine, motor and compressor). All other components of the Rankine engine also are located in the outdoor unit.

The Rankine-cycle engine also may be used to generate electricity by uncoupling the compressor from the electric motor (employing the electric clutch) and driving the motor as a generator. The power generation capabilities of the Rankine-cycle subsystem are not large. At design conditions ($195^{o}$F solar inlet temperature and $75^{o}$F outdoor wet bulb temperature), the motor/generator will produce approximately 1200 watts of electricity. Of this, 800 watts are used internally within the outdoor unit to run the condenser fan, the condenser sump pump, the Rankine-cycle feed pump and the RC/AC controls. Thus, at design conditions the net output from the outdoor unit would be about 400 watts. The collector loop pumps required to support Rankine engine operation consume 360 watts. Thus, the net electrical power generation capabilities of the system are minimal. However, the power generation mode normally is used only as a preferred alternative to rejecting collected solar energy to the environment (purging). The purge mode requires 450 watts of electricity and thus consumes considerably more power than the generation mode, which consumes little or no electrical power.

The RC/AC performance parameters for each month of operation are listed in Table 2-7. The most solar energy was used by the Rankine during June, of which 94 percent was used for cooling. Over the operating period, 17 x $10^{6}$ Btu of solar energy were delivered to the Rankine, of which 88 percent was used for cooling. The balance was used to generate electricity.

TABLE 2-7. RANKINE CYCLE SUMMARY

| MONTH | SOLAR ENERGY USED | | | RANKINE ENGINE EFFICIENCY[a] | AUXILIARY ELECTRIC ENERGY | OPERATING ENERGY | | GENERATED ELECTRICAL ENERGY | COOLING PRODUCED | SOLAR FLUID TEMPERATURE |
|---|---|---|---|---|---|---|---|---|---|---|
| | TOTAL | COOLING | GENERATION | | | COOLING | GENERATION | | | |
| | ($10^6$ Btu) | ($10^6$ Btu) | ($10^6$ Btu) | (PERCENT) | [$10^6$ Btu(e)] | [$10^6$ Btu(e)] | [$10^6$ Btu(e)] | [$10^6$ Btu(e)] | ($10^6$ Btu) | ($^{o}$F) |
| JANUARY 1981 | 0.37 | 0 | 0.37 | 10.1 | 0 | 0.0 | 0.02 | 0.00 | 0 | 192 |
| FEBRUARY 1981 | 0 | 0 | 0 | -- | 0 | 0 | 0 | 0 | 0 | -- |
| MARCH 1981 | 0 | 0 | 0 | -- | 0 | 0 | 0 | 0 | 0 | -- |
| APRIL 1981 | 2.10 | 1.03 | 1.07 | 8.0 | 0.38 | 0.15 | 0.05 | 0.025 | 1.31 | 183 |
| MAY 1981 | 3.94 | 3.78 | 0.16 | 8.2 | 0.75 | 0.34 | 0.01 | 0.005 | 2.87 | 176 |
| JUNE 1981 | 5.49 | 5.15 | 0.34 | 7.9 | 2.94 | 0.96 | 0.02 | 0.003 | 7.96 | 182 |
| JULY 1981 | (b) | (b) | (b) | (b) | (b) | (b) | (b) | (b) | (b) | (b) |
| AUGUST 1981 | 1.59 | 1.59 | 0 | 7.2 | 2.59 | 0.77 | 0 | 0 | 4.71 | 170 |
| TOTAL | 16.97 | 15.03 | 1.94 | -- | 9.29 | 2.96 | 0.12 | 0.042 | 22.79 | -- |
| AVERAGE[c] | 3.39 | 3.01 | 0.39 | -- | 1.86 | 0.59 | 0.02 | 0.01 | 4.56 | -- |

(a) Obtained from a look-up table using solar fluid inlet temperature to Rankine and condenser water temperature.
(b) Not available
(c) Seasonal average based and available data from April through August

The Rankine engine efficiencies listed in Table 2-7 were estimated from a lookup table as a function of solar fluid inlet temperature and condenser water temperature. The lookup table was derived from test data obtained under laboratory operating conditions and represents the thermal efficiency that could be expected under ideal operating conditions. The Shenandoah Rankine engine normally did not operate with a thermal efficiency as high as predicted during most of the OTP. Analysis of the data indicated that the thermal efficiency was as much as 30 to 40 percent lower than expected. The precise cause of the degraded Rankine engine performance is not known, but the main cause appears to be low boiler capacity. Other possible causes are air entrapment (the Freon 113 working fluid is normally subatmospheric), caused by leaks, and working fluid contamination.

The operating energies presented in Table 2-7 represent the electrical energy that was used to operate the solar loop pumps and to support the RC/AC "parasitics" (internal pumps, condenser fan and controls). The operating energy does not include the furnace fan, which was included in the cooling operating energy presented in Section 3.0. The operating energy was included in the table to provide complete information on the RC/AC performance characteristics.

# SECTION 3.0
# OPERATING ENERGY

For the Shenandoah solar heating and cooling system to provide domestic hot water and space conditioning, electrical energy was required to operate the pumps, fans, valves and controls within the system. Operating energy (system total or subsystem usage) is thus defined as the electrical energy used by the system to perform those functions that did not directly influence the thermal state of the system and consists of the energy used to power the pumps and fans within the system. The energy used by the system controls and the flow control valves was not instrumented because their power usage was very low.

Table 3-1 lists the operating energy consumed by the system and its subsystems during each month of the test period. The domestic hot water subystem required no operating energy. The energy collection and storage subsystem (ECSS) operating energy is the energy used by the system to collect and store solar energy when the Rankine engine is not operating. The space heating operating energy includes the furnace fan and solar heating pump energy usage. The space cooling operating energy consists of all energy used to support the cooling subsystem, including the furnace fan and the system pump P1 and is not the same as the values presented in Table 2-7, which are defined differently. The cooling subsystem operating energies are broken down into direct cooling mode, storage cooling mode and auxiliary mode. In the direct cooling mode, solar energy direct from collectors is used by Rankine engine to provide part of shaft power required by vapor compressors. The operating energy in this mode includes system pumps P1 and P2, R/C-A/C parasitics and fan energy. The operating energy in the storage cooling mode (energy to Rankine provided from storage) includes system pumps P1 and P2, storage pump P4 and R/C-A/C parasitics and fan energy. The operating energy in auxiliary cooling mode includes the A/C parasitics and fan energy. The total system operating energy includes all electrical energy consumed by the system

TABLE 3-1. SYSTEM OPERATING ENERGY

| MONTH | ECSS OPERATING ENERGY [$10^6$ Btu(e)] | SPACE COOLING OPERATING ENERGY | | | | SPACE HEATING OPERATING ENERGY | | GENERATION OPERATING ENERGY [$10^6$ Btu(e)] | TOTAL OPERATING ENERGY | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | TOTAL [$10^6$ Btu(e)] | DIRECT COOLING MODE [$10^6$ Btu(e)] | STORAGE COOLING MODE [$10^6$ Btu(e)] | AUXILIARY COOLING MODE [$10^6$ Btu(e)] | TOTAL [$10^6$ Btu(e)] | SOLAR HEATING MODE [$10^6$ Btu(e)] | | TOTAL SYSTEM [$10^6$ Btu(e) | SOLAR ASSIST MODE [$10^6$ Btu(e)] | SOLAR SPECIFIC [$10^6$ Btu(e)] |
| January 1981 | 0.16 | 0 | 0 | 0 | 0 | 0.23 | 0.18 | 0.02 | 0.41 | 0.36 | 0.26 |
| February 1981 | 0.14 | 0 | 0 | 0 | 0 | 0.13 | 0.10 | 0 | 0.27 | 0.25 | 0.19 |
| March 1981 | 0.17 | 0 | 0 | 0 | 0 | 0.05 | 0.04 | 0 | 0.27 | 0.20 | 0.18 |
| April 1981 | 0.15 | 0.20 | 0.11 | 0 | 0.09 | 0 | 0 | 0.08 | 0.43 | 0.33 | 0.27 |
| May 1981 | 0.10 | 0.52 | 0.26 | 0.14 | 0.12 | 0 | 0 | 0.01 | 0.63 | 0.51 | 0.29 |
| JUNE 1981 | 0.11 | 1.30 | 0.49 | 0.03 | 0.78 | 0 | 0 | 0.02 | 1.43 | 0.61 | 0.35 |
| JULY 1981 | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) |
| AUGUST 1981 | 0.14 | 1.06 | 0.17 | 0.04 | 0.85 | 0 | 0 | 0 | 1.19 | 0.35 | 0.22 |
| TOTAL | 0.97 | 3.08 | 1.03 | 0.21 | 1.84 | 0.41 | 0.32 | 0.13 | 4.58 | 2.64 | 1.76 |
| AVERAGE | 0.14 | 0.44 | 0.15 | 0.03 | 0.26 | 0.06 | 0.05 | 0.02 | 0.15 | 0.38 | 0.25 |

(a) Not available

except that used for auxiliary energy. It includes power generation and purge fan operating energies but is not credited for power generation, which appears only in electrical energy savings in the next section.

The total operating energy in the solar assist mode includes operating energies in direct cooling mode, storage cooling mode, solar heating mode, electric energy generation mode and ECSS operating energy. The solar specific operating energy includes operating energy used by system pumps P1 and P2, storage pump P3, heating pump P4, purge fan and part of R/C-A/C parasitics (calculated from difference in R/C-A/C parasitics in Rankine assist mode and auxiliary mode).

The subsystem operating energy by mode presented in this section can be combined with the contribution to load by mode presented in Section 2 for each subsystem, to provide a subsystem performance measure by mode and also to calculate actual energy savings which is presented later in Section 4.

Over the test period, the system used $4.58 \times 10^6$ Btu of electrical (operating) energy to meet the system load of $37.38 \times 10^6$ Btu ($9.38 \times 10^6$ Btu of DHW load, $11.15 \times 10^6$ Btu of space heating load and $16.85 \times 10^6$ Btu of space cooling load). This operating energy was used to support the use of both solar and auxiliary energy. The system used $0.97 \times 10^6$ Btu of ECSS operating energy to collect and store $35.6 \times 10^6$ Btu of solar energy. Therefore, 36.7 Btu of solar energy were collected and charged into thermal storage for each Btu of electrical energy used to support collection and storage.

Some of the performance measures which may be typical of such single family heating and cooling solar system were computed and are given in table 3-2. Notice, to compute the solar specific operating energy for DHW and space heating subsystems the ECSS operating energy was prorated using the ratio of the energy used by each subsystem to the total energy from storage.
Also notice that in the storage cooling mode a Btu of solar specific operating electric energy is required to collect, store and deliver 9 Btu of thermal

TABLE 3-2. SUBSYSTEM SOLAR ENERGY USE TO OPERATING ENERGY RATIO

| MODE OR SUBSYSTEM | RATIO,[a] $\frac{\text{Btu (th)}}{\text{Btu (e)}}$ |
|---|---|
| DIRECT COOLING | 23 |
| STORAGE COOLING | 9 |
| SOLAR HEATING | 18 |
| SOLAR TO DHW | 17 |
| SOLAR TO LOADS | 16[b] |

(a) $$\text{RATIO} = \frac{\text{SOLAR ENERGY USED BY SUBSYSTEM OR MODE}}{\text{SOLAR SPECIFIC OPERATING ENERGY FOR THE SUBSYSTEM OR MODE}}$$

(b) FOR EXAMPLE; SOLAR ENERGY DELIVERED TO LOAD SUBSYSTEMS = 28.34 MMBtu

SOLAR SPECIFIC OPERATING ENERGY (TABLE 3-1) = 1.76 MMBtu (e)

RATIO = 28.34/1.76 = 16

energy to Rankine subsystem. Therefore, in order for cooling from storage to break even the Rankine engine efficiency should be about 8 percent (= $\frac{1}{9}$ x 0.73; 0.73 is motor efficiency).

Utilizing the cooling load and auxiliary energy consumption in auxiliary mode from table 2-4 and auxiliary cooling operating energy from table 3-1, a coefficient of performance in the auxiliary mode is calculated to be 1.67. Similarly for the total cooling subsystem the coefficient of performance is 1.73.

# SECTION 4.0
# ENERGY SAVINGS

Whenever solar energy is used instead of or in addition to auxiliary energy, energy savings are realized. The energy saved by the Shenandoah solar heating and cooling system was determined by comparison of the energy used by the system with the energy that would have been required by assumed conventional load subsystems. The assumed conventional subsystems are the system's auxiliary load subsystems (gas hot water heater, gas fired furnace and motor-driven air conditioner), which are capable of meeting the entire house load. Energy used by components common to the solar and conventional subsystems (such as the furnace fan and the air conditioner condenser) was ignored because of their commonality. The energy used to support the solar collection, storage and delivery was necessarily debited against the energy savings. The power generated was credited to the energy savings. Energy savings were both electrical and fossil.

The electrical energy savings and fossil energy savings for the Shenandoah solar system for the OTP are listed in table 4-1. These are calculated as follows:

$$\text{(Electrical Energy Savings)} = \underbrace{\frac{\text{(Solar energy used for cooling)} \times \text{(Rankine Engine Efficiency)}}{\text{(Motor efficiency=0.73)}}}_{\text{Space cooling gross electrical energy savings}} + \text{(Energy Generated by Rankine Subsystem)} - \text{(Solar Specific Operating Energy)}$$

$$\text{(Fossil Energy Savings)} = \frac{\text{(Solar Energy Used for Space Heating)}}{\text{(Furnace Efficiency = 0.686)}} + \frac{\text{(Solar Energy used for DHW)}}{\text{(DHW heater efficiency =0.6)}}$$

TABLE 4-1. ENERGY SAVINGS

| MONTH | GROSS ELECTRICAL ENERGY SAVINGS | | | FOSSIL FUEL ENERGY SAVINGS | | | SOLAR SPECIFIC OPERATING ENERGY | NET ELECTRICAL ENERGY SAVINGS | | NET FOSSIL FUEL ENERGY SAVINGS |
|---|---|---|---|---|---|---|---|---|---|---|
| | TOTAL | SPACE COOLING | ENERGY GENERATION | TOTAL | SPACE HEATING | DHW | | | | |
| | [$10^6$ Btu(e)] | [$10^6$ Btu(e)] | [$10^6$ Btu(e)] | ($10^6$ Btu) | ($10^6$ Btu) | ($10^6$ Btu) | [$10^6$ Btu(e)] | [$10^6$ Btu(e)] | [kWh(e)] | [$10^6$ Btu] |
| JANUARY 1981 | 0.01 | 0 | 0.01 | 6.78 | 6.31 | 0.46 | 0.26 | -0.25 | -73 | 6.78 |
| FEBRUARY 1981 | 0 | 0 | 0 | 4.08 | 2.95 | 1.13 | 0.19 | -0.19 | -55 | 4.08 |
| MARCH 1981 | 0 | 0 | 0 | 5.67 | 3.32 | 2.34 | 0.18 | -0.18 | -53 | 5.67 |
| APRIL 1981 | 0.13 | 0.10 | 0.03 | 2.24 | 0 | 2.24 | 0.27 | -0.14 | -41 | 2.24 |
| MAY 1981 | 0.44 | 0.43 | 0.01 | 2.37 | 0 | 2.37 | 0.29 | +0.15 | +44 | 2.37 |
| JUNE 1981 | 0.56 | 0.56 | 0 | 2.27 | 0 | 2.27 | 0.35 | +0.21 | +62 | 2.27 |
| JULY 1981 | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) | (a) |
| AUGUST 1981 | 0.16 | 0.16 | 0 | 2.72 | 0 | 2.72 | 0.22 | -0.06 | -18 | 2.72 |
| TOTAL | 1.30 | 1.25 | 0.05 | 23.99 | 12.58 | 11.40 | 1.76 | -0.40 | -134 | 26.13 |
| AVERAGE | 0.19 | 0.18 | 0.01 | 3.43 | 1.80 | 1.63 | 0.25 | -0.06 | -19 | 3.73 |

(a) Not Available

The fossil energy used in the auxiliary mode was natural gas. To compute the cubic feet of natural gas savings divided fossil energy savings by the heating value of natural gas (= 1014 Btu/ft$^3$). Over the 7 month of the test period the net fossil energy savings were 26.13 MMBtu (= 25770 cubic feet of natural gas) while the net electrical energy savings were a loss of 0.4 MMBtu or 116 kWh. It should be noted that electrical energy was required as operating energy to support space heating and DHW subsystems.

The monetary savings to the homeowner for the test period is calculated to be $122. For calculating monetary savings the cost of natural gas was assumed to be $0.50 per hundred cubic feet and the cost of electricity was assumed to be $0.06 per kWh.

On an annual basis the subsystem loads during OTP represent about 50 percent of the space heating load, 60 percent of the space cooling load and 58 percent of the DHW load. Utilizing these factors the net annual electrical savings are projected to be a loss of 280 kWh and the net annual fossil fuel savings are projected to be 50 MMBtu (49300 cubic feet of natural gas). The monetary savings are projected to be $230 per year.

Computing the gross electrical energy savings for cooling subsystem, Rankine engine efficiencies are required. The Rankine engine efficiencies used were from a look up table which utilizes solar inlet temperature to the Rankine and condenser water temperature. The look up table was prepared under laboratory conditions. However, the actual Rankine engine efficiency was significantly lower than that from the look up table. Therefore, the electrical energy savings may be overestimated.

Alternate Method

An alternate method for computing gross cooling electrical energy savings is by comparing the coefficient of performance of the air-conditioner in the auxiliary mode with that of the total system in Rankine assist mode. These are calculated using the values from Table 2-6 as follows:

Auxiliary Mode (Conventional)

Cooling Load = $11.78 \times 10^6$ Btu

Auxiliary Energy = $5.20 \times 10^6$ Btu(e)

"COP" $= \dfrac{11.78 \times 10^6}{5.2 \times 10^6} = 2.265$

Total System (with Solar)

Cooling Load = $16.85 \times 10^6$, Btu

Auxiliary Energy = $6.66 \times 10^6$, Btu

"COP" $= \dfrac{16.85 \times 10^6}{6.66 \times 10^6} = 2.530$

Gross Cooling Electrical Energy Savings $= \dfrac{1}{2.265} \quad \dfrac{1}{2.530} \times 16.85 \times 10^6$

$= 0.78 \times 10^6$, Btu(e)

Solar Specific Operating Energy (Table 3-1) = $1.76 \times 10^6$, Btu(e)

Net Electrical Energy Savings for OTP = $(0.78 - 1.76)\ 10^6 = -0.98 \times 10^6$, Btu(e) [-287 kWh(e)]

The fossil savings are the same as before i.e., 26.13 MMBtu. The monetary savings for the OTP is $112. The annual monetary savings is projected to be about $215.

# SECTION 5

# MAINTENANCE

Most of the components in the Shenandoah system were off-the-shelf parts and were maintenance free. However, both the Rankine-cycle air conditioner (RC/AC) and microprocessor-based Tailorable Control Panel (TCP), being propotypes, experienced some problems.

The Gearbox between the Rankine-cycle turbine and the motor/generator failed in July, 1981. The gearbox was replaced at the beginning of August, 1981. This was the only major problem with the Shenandoah RC/AC during the Operational Test Period.

In February, 1981, a faulty relay in the TCP hampered the performance of the system when it was in the heating mode. After the relay was replaced the system functioned properly.

In July, 1981 several short power failures resulted in malfunctioning of the TCP. After a power failure the TCP would normally reset itself and go through a check-out sequence before resuming system operation. However, if the power failure is of extremely short duration the TCP would not reset itself but would begin operation at an arbitrary point in its operating software resulting in system shutdown. This problem has been resolved.

# SECTION 6.0

# SUMMARY AND CONCLUSIONS

This system performance evaluation provides an operational summary, over the test period (January 1981 through August 1981), of a single family residential solar heating and cooling system located in Shenandoah (Newnan), Georgia. The system (designed by Honeywell Inc.) consisted of a 702 square feet of double glazed flat plate solar collectors (manufactured by Lennox Industries), a 1000 gallon thermal storage tank to provide thermal energy primarily for space heaitng and domsetic hot water, a developmental organic Rankine engine (developed by Barber-Nichols) to provide part of shaft power to a vapor compressor cooling subsystem and a microprocessor based system control panel (developed by Honeywell Inc.) to provide for automatic solar system operation and diagnostics. This analysis was conducted by evaluation of measured system performance (Site Data Acquisition and processing done by Vitro Laboratories) and by comparison of measured climatic data with long term average climatic condition for the site.

The following observations were made:

1. Measured average daily insolation and average temperature for the Operational Test Period (OTP) was close to the long term average weather conditions. However, during winter the insolation was about 20 percent higher and the average ambient temperature slightly higher than long term averages. The result was a lower heating load and a higher than expected solar contribution. During spring and summer the insolation was about 17 percent lower than the long term average. This partly contributed to the lower than expected solar contribution for space cooling.

2. The solar system provided energy to the building space heat, space cool and hot water loads during the OTP, providing 77 percent of the space heating, 15 percent of the space cooling and 76 percent of the domestic hot water (DHW) energy. The result was a net fossil energy savings of about 26 million Btu or 23000 cubic feet of natural gas at the expense

of 117 kWh of electrical energy for the OTP. The projected annual savings are 50 million Btu of fossil energy (49300 cubic feet of natural gas) at the expense of 280 kWh of electrical energy. The monetary savings with 1981 costs are expected to be $230 per year.

3. All the solar system components were state-of-the-art except the Rankine engine which was a developmental unit. The Rankine engine efficiencies could not be measured directly because there was no provision for output shaft power measurements. However, comparison of auxiliary energy usage by electric motor during solar cooling (Rankine assist mode) with auxiliary cooling (Rankine engine not operational) suggested a lower than expected Rankine engine efficiency. The lower Rankine engine performance was attributed to lower than designed boiler capacity, air entrapment caused by leaks (the Freon 113 working fluid is normally subatmospheric) and working fluid contamination. In the larger 25-ton Rankine engines built subsequent to this installation an automatic air purge system was incorporated. This signifcantly improved seasonal performance.

4. The DHW subsystem at Shenandoah performed exceptionally well especially after the storage charge setpoint temperature for winter was increased from $140^{o}$F to $190^{o}$F and the DHW heater temperature limit was lowered from $170^{o}$F to $140^{o}$F. The result was a 90 percent solar contribution due to the change of storage setpoints and decreased standby losses due to change of DHW temperature limit. The annual fossil fuel saved due to solar contribution to DHW was about 18 MMBtu or 17500 cubic feet of natural gas. Savings in fossil energy, for such DHW and space heating subsystems, are always associated with an inherent entropy conservation.

5. The option of cooling from storage is uneconomical, due largely to the large operating energy (electrical) required to collect, store and deliver low grade thermal energy (see Table 3-2) for shaft power generation in Rankine engine. However, in the Shenandoah system when solar was available and there was no cooling load, storage was charged

(to provide for cooling during non-solar hours) in lieu of available excess heat purging or power generation. Purging excess heat consumes more electrical power than cooling from storage. Therefore, storage cooling is a better option than purging. However, the option for power generation during summer with another smaller storage tank to provide for DHW should be studied further.

# SECTION 7.0
# REFERENCES

[1] Cinquemani, V., et al. "Input Data for Solar Systems," prepared for t U.S. Department of Energy by the National Climatic Center, Asheville, 1978.

[2] "Long-Term Weathering Effects on the Thermal Performance of the Lennox Honeywell (Liquid) Solar Collectors," prepared for the U.S. Departmen of Energy by the Wyle Laboratories, Huntsville, Alabama, 1978, Dept. No DOE/NASA CR-150818.

[3] Streed, E., et al. "Thermal Data Requirements and Performance Evaluatio Procedures for the National Solar Heating and Cooling Demonstration Program," prepared for the Energy Research and Development Administration by the Office of Housing and Building Technology and Thermal Engineering Section, Washington, DC, August 1976, NBSIR 76-1137.

# APPENDIX A

## DEFINITION OF PERFORMANCE FACTORS AND SOLAR TERMS

### COLLECTOR ARRAY PERFORMANCE

The collector array performance is characterized by the amount of solar energy collected with respect to the energy available to be collected.

- INCIDENT SOLAR ENERGY (SEA) is the total insolation available on the gross collector array area. This is the area of the collector array energy-receiving aperture, including the framework which is an integral part of the collector structure.

- OPERATIONAL INCIDENT ENERGY (SEOP) is the amount of solar energy incident on the collector array during the time that the collector loop is active (attempting to collect energy).

- COLLECTED SOLAR ENERGY (SECA) is the thermal energy removed from the collector array by the energy transport medium.

- COLLECTOR ARRAY EFFICIENCY (CAREF) is the ratio of the energy collected to the total solar energy incident on the collector array. It should be emphasized that this efficiency factor is for the collector array, and available energy includes the energy incident on the array when the collector loop is inactive. This efficiency must not be confused with the more common collector efficiency figures which are determined from instantaneous test data obtained during steady state operation of a single collector unit. These efficiency figures are often provided by collector manufacturers or presented in technical journals to characterize the functional capability of a particular collector design. In general, the collector panel maximum efficiency factor will be significantly higher than the collector array efficiency reported here.

## ENERGY COLLECTION AND STORAGE SUBSYSTEM

The Energy Collection and Storage Subsystem (ECSS) is composed of the collector array, the primary storage medium, the transport loops between these, and other components in the system design which are necessary to mechanize the collector and storage equipment.

- INCIDENT SOLAR ENERGY (SEA) is the total insolation available on the gross collector array area. This is the area of the collector array energy-receiving aperture, including the framework which is an integral part of the collector structure.

- AMBIENT TEMPERATURE (TA) is the average temperature of the outdoor environment at the site.

- ENERGY TO LOADS (SEL) is the total thermal energy transported from the ECSS to all load subsystems.

- ECSS OPERATING ENERGY (CSOPE) is the critical operating energy required to support the ECSS heat transfer loops.

- ECSS REJECTED ENERGY (CSRJE) is the energy intentionally rejected by the purge unit as a solar fluid overtemperature protection and to prevent damage to the system components.

- ECSS SOLAR CONVERSION EFFICIENCY (CSCEF) is the ratio of the solar energy delivered to the load subsystems to the incident solar energy.

## SPACE HEATING SUBSYSTEM

The space heating subsystem is characterized by performance factors accounting for the complete energy flow to and from the subsystem. The average building temperature and the average ambient temperature are tabulated to indicate the relative performance of the subsystem in satisfying the space heating load and in controlling the temperature of the conditioned space.

## ENVIRONMENTAL SUMMARY

The environmental summary is a collection of the weather data which is generally instrumented at each site in the Development Program. It is tabulated in this report for two purposes (1) as a measure of the conditions prevalent during the operation of the system at the site, and (2) as a historical record of weather data for the vicinity of the site.

- TOTAL INSOLATION (SE) is the accumulated total solar energy incident upon the gross collector array (per unit area) measured at the site.

- AMBIENT TEMPERATURE (TA) is the average temperature of the environment at the site.

- DAYTIME AMBIENT TEMPERATURE (TDA) is the ambient temperature during the period from three hours before solar noon to three hours after solar noon.

## SPACE COOLING SUBSYSTEM

The space cooling subsystem is characterized by performance factors accounting for the complete energy flow to and from the subsystem. The average building temperature and the average ambient temperature are tabulated to indicate the relative performance of the subsystem in satisfying the space cooling load and in controlling the temperature of the conditioned space.

- SPACE-COOLING LOAD (CL) is the amount of thermal energy removed from the conditioned space of the building by all central SCS solar-and auxiliary-powered equipment.

- SOLAR FRACTION OF LOAD (CSFR) is the percentage of the SCS load demand which is supplied from the air conditioners which is attributable to the solar-powered Rankine cycle engine.

- SOLAR ENERGY USED (CSE) is the amount of solar energy supplied from the ECSS to the space cooling subsystem which is used to satisfy the space cooling load.

- OPERATING ENERGY (COPE) is teh amount of electrical energy required to support the operation of the space cooling subsystem (e.g., fans, pumps, etc.) which is not intended to directly affect the thermal state of the subsystem.

- AUXILIARY THERMAL USED (CAT) is the equivalent thermal energy supplied to the space cooling subsystem by the subsystem auxiliary equipment.

- AUXILIARY ELECTRICAL FUEL (CAE) is the amount of electrical energy supplied to the space cooling subsystem auxiliary equipment.

- ELECTRICAL ENERGY SAVINGS (CSVE) is the estimated different between the electrical energy requirements of a conventional space cooling system (carrying the full cooling load) and the electrical energy required by the actual solar-assisted subsystem.

- BUILDING TEMPERATURE (TB) is the average conditioned space dry bulb temperature.

- AMBIENT TEMPERATURE (TA) is the average ambient dry bulb temperature at the site.

- DAYTIME AMBIENT TEMPERATURE (TDA) is the ambient temperature during the period from three hours before solar noon to three hours after solar noon.

RANKINE CYCLE SUMMARY

The Rankine cycle summary is a combination of performance factors which outline the performance of the Rankine Cycle Air Conditioner during space cooling and power generation.

# APPENDIX B

## SOLAR ENERGY SYSTEM PERFORMANCE EQUATIONS FOR SHENANDOAH, GEORGIA (OTS #41)

### I. INTRODUCTION

Solar energy system performance is evaluated by performing energy balance calculations on the system and its major subsystems. These calculations are based on physical measurement data taken from each subsystem every 320 seconds. This data is then numerically combined to determine the hourly, daily, and monthly performance of the system. This appendix describes the general computational methods and the specific energy balance equations used for this evaluation.

Data samples from the system measurements are numerically integrated to provide discrete approximations of the continuous functions which characterize the system's dynamic behavior. This numerical integration is performed by summation over the total time period of interest of the product of the measured rate of the appropriate performance parameters and the sampling interval.

There are several general forms of numerical integration equations which are applied to each site. These general forms are exemplified as follows: The total solar energy available to the collector array is given by

$$\text{SOLAR ENERGY AVAILABLE} = (1/60) \times \Sigma[\text{I001} \times \text{CLAREA}] \times \Delta\tau$$

where I001 is the solar radiation measurement provided by the pyranometer in Btu/ft$^2$-hr, CLAREA is the area of the collector array in square feet, $\Delta\tau$ is the sampling interval in minutes, and the factor (1/60) is included to correct the solar radiation "rate" to the proper units of time.

These equations are comparable to those specified in "Thermal Data Requirements and Performance Evaluation Procedures for the National Solar Heating and Cooling Demonstration Program." This document was prepared by an inter-agency committee of the government, and presents guidelines for thermal performance evaluation.

Performance factors are computed for each hour of the day. Each numerical integration process, therefore, is performed over a period of one hour. Since long-term performance data is desired, it is necessary to build these hourly performance factors to daily values. This is accomplished, for energy parameters, by summing the 24 hourly values. For temperatures, the hourly values are averaged. Certain special factors, such as efficiencies, require appropriate handling to properly weight each hourly sample for the daily value cmputation. Similar procedures are required to convert daily values to monthly values.

## II. PERFORMANCE EQUATIONS

The performance equations for the Honeywell Thomas Village heating and cooling system used for the data evaluation of this report are contained in the following pages and have been included for technical reference and information.

# EQUATIONS USED IN MONTHLY PERFORMANCE ASSESSMENT

Note: Measurement numbers reference system schematic, Figure C-1.

**Average Ambient Temperature ($^{o}$F)**

$$TA = (1/60) \times \Sigma T001 \times \Delta T$$

**Average Building Temperature ($^{o}$F)**

$$TB = (1/60) \times \Sigma T600 \times \Delta T$$

**Daytime Average Ambient Temperature**

$$TDA = (1/360) \times \Sigma T001 \times \Delta T$$

for $\pm$ 3 hours from solar noon

**Incident Solar Energy per Square Foot (Btu/ft$^2$)**

$$SE = (1/60) \times \Sigma I001 \times \Delta T$$

**Incident Solar Energy (Btu)**

$$SEA = (1/60) \times \Sigma[I001 \times CLAREA] \times \Delta T$$

**Operational Incident Solar (Btu)**

$$SEOP = (1/60) \times \Sigma[I001 \times CLAREA] \times \Delta T$$

when the collector loop is active

**Solar Energy Collected by the Array (Btu)**

$$SECA = \Sigma[M200 \times Cp \times (T101 - T100] \times \Delta T$$

**Energy Rejected by Collector Array (Btu)**

$$CSRJE = \Sigma[M100 \times Cp \times (T101 - T150)] \times \Delta T$$

Enthalpy Function for Water (BTU/LBM)

$$HWD\ (T_2, T_1) = \int_{T_1}^{T_2} C_p\ (T)dt$$

This function computes the enthalpy change of water as it passes through a heat exchanging device.

Solar Energy to Storage (BTU)

$$STEI = \Sigma[M200 \times HWD\ (T250,\ T200] \times \Delta T$$

whenever energy is delivered to storage

Solar Energy from Storage (Btu)

$$STEO = \Sigma[M200 \times HWD\ (T200,\ T250) + M400 \times HWD\ (T400,\ T450) + M300 \times HWD\ (T350,\ T300)] \times \Delta T$$

Average Temperature of Storage ($^{o}F$)

$$TST = (1/60)\ \Sigma[(T201 + T202 + T203)/3] \times \Delta T$$

Hot Water Auxiliary Energy (Btu)

$$HWAF = 1014 \times F300$$

$$HWAT = 0.6 \times HWAF$$

Solar Energy Used for Hot Water (Btu)

$$HWSE = \Sigma[M300 \times HWD\ (T350,\ T300)] \times \Delta T$$

Hot Water Consumed (gallons)

$$HWCSM = \Sigma[W300] \times \Delta T$$

Hot Water Load (Btu)

$$HWL = \Sigma[M300 \times HWD\ (T351,\ T300)] \times \Delta T$$

Hot Water Standby Losses (Btu)

$$HWLOSS = HWSE + HWAT - HWL$$

Heating Auxiliary Energy (Btu)

$$HAF = 1014 \times F400$$
$$HAT = 0.686 \times HAF$$

Solar Energy Used for Heating (Btu)

$$HSE = \Sigma[M400 \times HWD\ (T400,\ T450)] \times \Delta T$$

Heating Load (Btu)

$$HL = HSE + HAT$$

Heating Solar Fraction (percent)

$$HSFR = 100 \times HSE/HL$$

Cooling Load (Btu)

$$CL = \Sigma[M600 \times \Delta H_a\ (T601,\ TH501,\ T603,\ RH502)] \times \Delta T$$

Solar Energy Used for Cooling (Btu)

$$CSE = \Sigma[M100 \times Cp \times (T550 - T151)] \times \Delta T$$

calculated only when cooling.

Auxiliary Electric Used for Cooling (Btu)

$$CAE = (3413/60) \times \Sigma[EP503] \times \Delta T$$

Solar Energy Delivered to Rankine (Btu)

$$RSE = \Sigma[M100 \times Cp \times (T550 - T151)] \times \Delta T$$

calculated whenever the Rankine engine is running

Rankine Efficiency

REFF is determined from a look-up table as a function of T151 and T550.

Solar Heating Operating Energy (Btu)

$$HOPE1 = (3413/60) \times \Sigma[EP400] \times \Delta T$$

Power Generated (Btu)

$$PWRGEN = (3413/60) \times \Sigma[EP504] \times \Delta T$$

Cooling Solar Fraction (percent)

$$CSFR = 100 \times (CSE \times REFF)/(CSE \times REFF + CAE \times 0.73)$$

Rankine Average Solar Fluid Supply Temperature ($^{o}F$)

$$TRANKS = \{\Sigma[M100 \times T151] \times \Delta T\} / \{\Sigma[M100] \times \Delta T\}$$

whenever the Rankine is operating

Rankine Average Condensing Water TEmperature ($^{o}F$)

$$TRANKC = \{\Sigma[M100 \times T552] \times \Delta T\} / \{\Sigma[M100] \times \Delta T\}$$

whenever the Rankine is operating

Rankine Cycle Air Conditioner Parasitics (Btu)

$$PARA = (3413/60) \times \Sigma[EP501 + EP504 - EP503 - EP502] \times \Delta T$$

ECSS Operating Engery (Btu)

$$CSOPE = (3413/60) \times \Sigma[EP100 + EP200 + EP101] \times \Delta T$$

whenever collector array is operating.

Space Heating Operating Energy (Btu)

$$HOPE = (3413/60) \times \Sigma[EP400 + EP401] \times \Delta T$$

Space Cooling Operating Energy (BTU)*

$$COPE = PARA + (3413/60) \times \Sigma[EP401 + EP100 + EP200 + EP500] \times \Delta T$$

Whenever cooling

Rankine Cycle Operating Energy (Btu)*

$$TCEOPE = PARA + (3413/60) \times \Sigma[EP100 + EP200 + EP500] \times \Delta T$$

whenever Rankine is operating.

* These operating energies are divided by mode and for power generation.

**System Operating Energy** (Btu)

$$SYSOPE = (3413/60) \times \Sigma[EP100 + EP101 + EP200 + EP400 + EP401 + EP500] \times \Delta T + PARA$$

**Hot Water Fossil Fuel Savings** (Btu)

$$HWSVF = HWSE/.6$$

**Space Heating Electrical Savings** (Btu)

$$HSVE = -HOPE1$$

**Total Net Electrical Savings** (Btu)

$$TSVE = HSVE + CSVE - CSOPE + PWRGEN - GENOPE$$

# APPENDIX C.

## SITE DATA ACQUISITION SUBSYSTEM (SDAS)

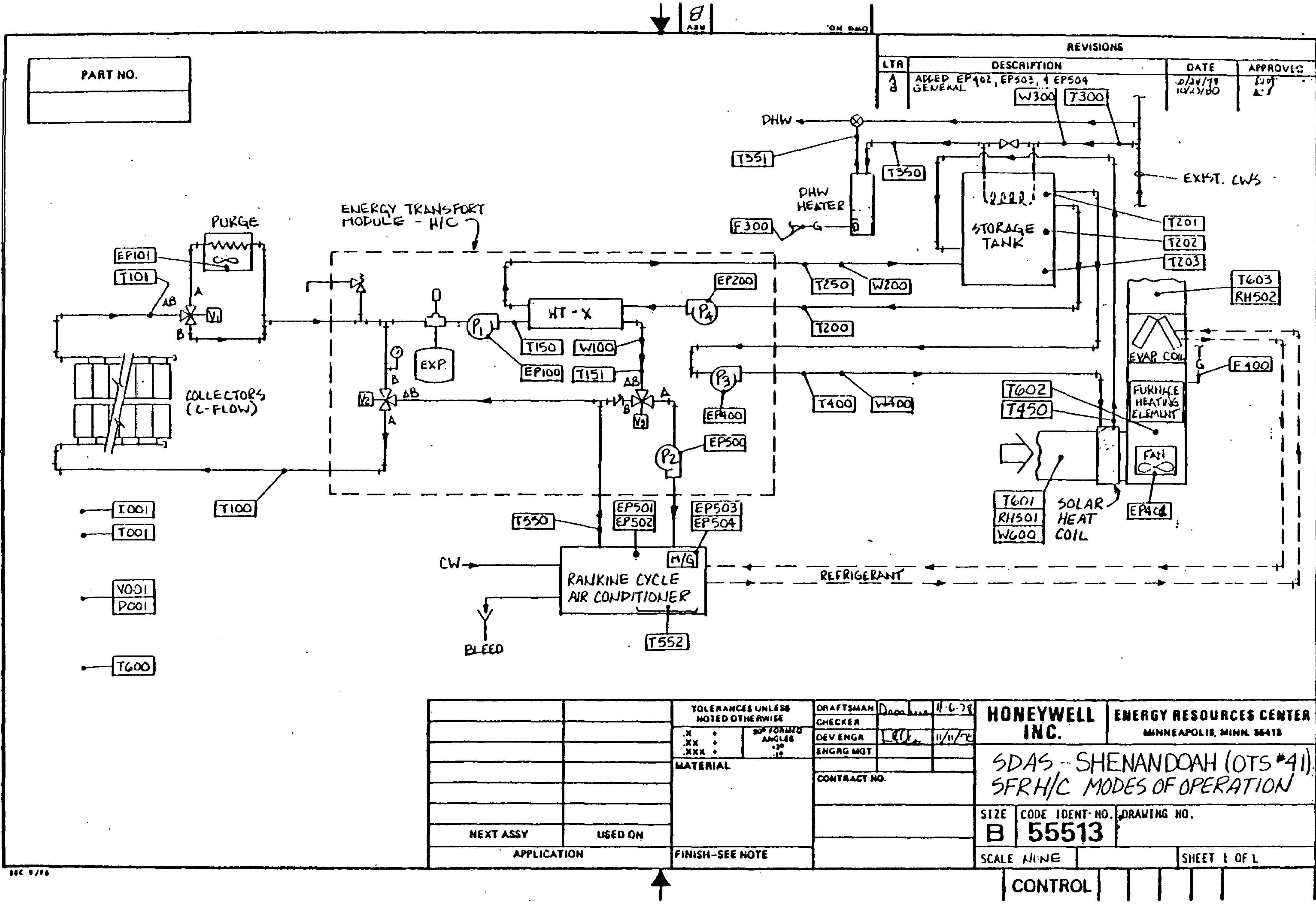
The Site Data Acquisition Subsystem (SDAS) consists of 43 sensors, an electrical junction box and an SDAS module. The descriptions and designations of all sensors are listed in Table C-1 and their locations are shown in Figure C-1. The module samples the sensor readings and stores the data on magnetic tape. Temperatures and electrical powers are sampled at 32 second intervals and 10 measurements are averaged to generate the values which are stored on tape at 320 second intervals. Flow rates are sampled only once each period, at the end of the 320 second interval, resulting in an instantaneous flow rate representing the flow for the whole period. The SDAS module transfers the stored data via telephone at the end of each day to a central computer operated by Vitro Engineering Corp. in Silver Springs, Maryland. The data are then stored on magnetic disc and/or magnetic tape.

At the end of each month, the data are reduced by the central computer using a program (called the site performance equations or Site Equation Document) which is unique to the site. Hourly, daily and monthly values for each performance parameter are calculated and printed. These reduced data were used to write this report. All raw data are archived for future reference.

| Unit | Description | Probe/Part No. | Thermowell No. |
|---|---|---|---|
| T001 | Outdoor Ambient Temp. | S53-P40Z36 | IS4 |
| T100 | Collector Inlet Temperature | S53-P40Z36 | F203U6 |
| T150 | HX Inlet (Collector Loop) Collector Outlet Temp. | S53-P40Z36 | F203U6 |
| T151 | HX Outlet (Collector Loop) Temperature | S53-P40Z36 | F203U6 |
| T101 | Purge Inlet Temperature | S53-P40Z36 | F203U6 |
| T200 | HX Inlet (Storage Loop) Temperature | S53-P40Z36 | F203U6 |
| T250 | HX Outlet (Storage Loop) Temperature | S53-P40Z36 | F203U6 |
| T201 | Storage Tank Temperature - Top | S53-P180Z36 | F203U154 |
| T202 | Storage Tank Temperature - Middle | S53-P180Z36 | F203U154 |
| T203 | Storage Tank Temperature - Bottom | S53-P180Z36 | F203U154 |
| T300 | Domestic Cold Water Temp. | S53-P30Z36 | F203U4 |
| T350 | DHW Solar Preheat Coil Outlet Temperature | S53-P30Z36 | F203U4 |
| T351 | DHW Temperature | S53-P100Z36 | F203U70 |
| T400 | Heating Coil (Solar) Inlet Temperature | S53-P40Z36 | F203U6 |
| T450 | Heating Coil (Solar) Outlet Temperature | S53-P40Z36 | F203U6 |
| T550 | Rankine Engine Solar Outlet Temperature | S53-P40Z36 | F203U6 |
| T552 | R/C Condensor Temp. | S53-P40Z36 | XF203U15 |
| T600 | Room Ambient | S7850 | Mount Furnished |
| T601 | House Return Air Temp. | S53-P80Z36 | F132 |
| T602 | Solar Heat Outlet Temp. | S53-P80Z36 | F132 |
| T603 | House Supply Air Temp. | S53-P80Z36 | F132 |
| T700 | J-Box Temp. | S53-P40Z36 | |
| T701 | SDAS Ambient Temp. | S53-P40Z36 | |
| W100 | Collector Loop Flow Rate | MKV-1-1/4-J07 | |
| W200 | Storage Loop Flow Rate | MKV-1-1/4-J07 | |
| W300 | DHW Flow Rate | 7845124-3 | |
| W400 | Heating Loop Flow Rate | MKV-1-1/4-J07 | |
| W600 | House Return Air Flow Rate | 430-2 | F132 |
| F300 | DHW Heater Gas Flow Rate | AL 85 | |
| F400 | Auxiliary Furnace Gas Flow Rate | AL 85 | |
| EP100 | Collector Loop Pump P1 Power | PC5-1F | |
| EP101 | Purge Fan Power | PC5-1F | |
| EP200 | Storage Loop Pump P4 Power | PC5-1F | |
| EP400 | Heating Loop Pump P3 Power | PC5-1F | |
| EP401 | Furnace Fan Power | PC5-1F | |
| EP500 | Rankine Solar Booster Pump P2 Power | PC5-1F | |
| EP501 | Rankine Unit Net Power | PC5-52F | |
| EP502 | Rankine Unit Output Power | See Comments | |
| EP503 | Rankine Motor Power | PC5-29F | |
| EP504 | Rankine Generator Power Output | See Comments | |
| I001 | Pyranometer | PSP | |
| RH501 | House Return Air Relative Humidity | HM-14-U | |
| RH502 | House Supply Air Relative Humidity | HM-14-U | |
| | SDAS Module | 7834400-3 | |
| | J-Box & Cables | 7833650 | |



T/ C- SD ENS IST




C-3

FIGURE C-1. SDAS SCHEMATIC

# APPENDIX D

## UNCERTAINTY ANALYSIS OF SOLAR PERFORMANCE FACTORS

The uncertainty of determining the performance evaluation factors for a particular solar energy system is related to the data requirement accuracy for sensor signal conditioning, the data acquisition sampling rate and the data processing method [3]. An error analysis of the calculations of certain performance parameters, based only on sensor accuracies, was conducted and is presented here. Two methods in general use for combining precision errors in measuring several variables to estimate the error in the calculated performance factors are (1) absolute limits and (2) statistical bounds. The derivation and use of these methods is given in Reference [3]. The results of this analysis for Shenandoah are presented in Table D-1. Notice that the measurement accuracy for the cooling load calculation was ± 18 percent. This happens because two discrete resistance bulbs are used to measure a small temperature drop across the chiller as opposed to using differential thermocouples on bridge-connected resistance bulbs.

Table D-1. Uncertainty Analysis Results

| PERFORMANCE FACTOR | TOLERANCE OR UNCERTAINTY (PERCENT) |
|---|---|
| SOLAR COLLECTOR EFFICIENCY | ± 10.8 |
| HEAT RATE TO RANKINE BOILERS | ± 7.7 |
| RANKINE GENERATED SHAFT POWER | ± 9.2 |
| (RANKINE + AUXILIARY) POWER | ± 5.2 |
| COOLING PRODUCED | ± 18.0 |
| BUILDING COOLING LOAD | ± 12.4 |
| FRACTION SOLAR CONTRIBUTION TO COOLING LOAD | ± 24 |
| THE FOLLOWING SENSOR ACCURACIES WERE ASSUMED FOR ESTIMATING THE UNCERTAINTIES: | |
| TEMPERATURE | ± 0.5°F |
| FLOW RATE | ± 3 PERCENT |
| POWER (EP SENSORS) | ± 2 PERCENT |
| RANKINE EFFICIENCY | ± 5 PERCENT |

# APPENDIX E

## LONG-TERM AVERAGE WEATHER CONDITIONS

The environmental estimates given in this appendix provide a point of reference for evaluation of weather conditions as reported in the Monthly Performance Assessments and Solar Energy System Performance Evaluations issued by the National Solar Data Program. As such, the information presented can be useful in prediction of long-term system performance.

Environmental estimates for this site include the following monthly averages: insolation on a horizontal plane at the site, insolation in the tilt plane of the collection surface, ambient temperature, heating degree-days, and cooling degree-days. Estimation procedures and data sources are detailed in the following paragraphs.

The preferred source of long-term temperature and insolation data is "Input Data for Solar Systems" (IDSS) [1] since this has been recognized as the solar standard. The IDSS data are used whenever possible in these environmental estimates for both insolation and temperature related sources; however, a secondary source used for insolation data is the Climatic Atlas of the United States , and for temperature related data, the secondary source is "Local Climatological Data".

Since the available long-term insolation data are only given for a horizontal surface, solar collection subsystem orientation information is used is an algorithm [4] to calculate the insolation expected in the tilt plane of the collector. This calculation is made using a ground reflectance of 0.2.

| MONTH | NORMAL MONTHLY TEMPERATURE (DEG F)* | NORMAL DEGREE DAYS* BASE 65 DEG F | | MEAN DAILY SOLAR RADIATION # (BTU/FT$^2$) | | |
|---|---|---|---|---|---|---|
| | | HEATING | COOLING | HORIZONTAL PLANE ($\bar{H}$) | COLLECTOR ARRAY PLANE ($\bar{H}_T$) | $\bar{H}_T/\bar{H}$ |
| JAN | 42.4 | 701 | 0 | 717.6 | 971.8 | 1.35 |
| FEB | 45.0 | 560 | 0 | 968.9 | 1205.7 | 1.24 |
| MAR | 51.1 | 443 | 12 | 1303.6 | 1463.8 | 1.12 |
| APR | 61.1 | 144 | 27 | 1686.2 | 1697.6 | 1.01 |
| MAY | 69.1 | 27 | 154 | 1853.8 | 1713.3 | 0.92 |
| JUN | 75.6 | 0 | 321 | 1913.8 | 1704.1 | 0.89 |
| JUL | 78.0 | 0 | 403 | 1812.2 | 1643.0 | 0.91 |
| AUG | 77.5 | 0 | 388 | 1708.5 | 1655.5 | 0.97 |
| SEP | 72.3 | 8 | 227 | 1422.0 | 1527.2 | 1.07 |
| OCT | 62.4 | 137 | 58 | 1199.9 | 1483.1 | 1.24 |
| NOV | 51.4 | 408 | 0 | 882.9 | 1215.9 | 1.38 |
| DEC | 43.5 | 667 | 0 | 674.2 | 946.6 | 1.40 |
| ANN | 60.8 | 3095 | 1589 | 1345.3 | 1463.3 | 1.07 |

* Based on 1941-1970 period
# As noted in Solmet Volume 1

ATLANTA, GEORGIA LATITUDE: 3339N LONGITUDE: 8426W

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